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For full announcement see list following index.

STEAM POWER

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PREFACE TO SECOND EDITION

THE comments and criticisms which have come to the authors during the five years since this book appeared do not seem to call for any radical change in arrangement or treatment. There does appear to be a demand for the addition of a small amount of material. This has been met by the inclusion of a chapter on " Performance of Steam Power Equipment " in which are included treatments of those subjects which users of the book seem to feel are necessary for completeness.

Other additions of minor character have been made to care for the development of the art since the date of the first edition.

The entire text has been carefully reviewed and certain parts have been rewritten to make them clearer. Certain minor errors and misprints discovered by users have also been corrected.

The authors desire to express their sincere thanks to all the users of this book who have assisted in this revision by indicating errors and omissions and points at which the text was not readily intelligible.

C. F. H.
T. C. U.

PREFACE TO FIRST EDITION

The following pages represent the results of an attempt to collect in a comparatively small book such parts of the field of steam power as should be familiar to engineers whose work does not require that they be conversant with the more complicated thermodynamic principles considered in advanced treatments. The experience of the authors has led them to believe that a book of this sort should give a correct view-point with regard to the use of heat in the power plant even though it does not enter deeply into the theoretical considerations leading up to that view-point; that it should supply the tools required for the solution of power plant problems of the common sort; and that it should give sufficient description of power plant apparatus to make the reader fairly familiar with the more common types.

Mathematical treatment of the subject has been eliminated to the greatest possible extent, and anyone familiar with elementary algebra should be able to understand readily such equations as it has been deemed necessary to include.

Brief explanations of physical and chemical concepts are given in every case in which the text required their use, so that those who have not studied these subjects, and those who have but have failed to crystallize and hold the neces-

sary ideas, should have little difficulty in reading the text understandingly.

It is hoped that the book may prove serviceable as a text for steam power courses given to civil engineers in the various colleges and that it may also meet the needs of those instructing power plant operators in industrial schools.

C. F. H.
T. C. U.

June, 1916.

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CHAPTER I

PHYSICAL CONCEPTIONS AND UNITS

1. Matter. The universe is generally pictured as composed of matter and energy. Matter is regarded as that which is possessed of mass, or as that which is possessed of inertia; i.e., which *requires the action of force to put it in motion, to bring it to rest or to change its velocity.* These definitions merely enumerate characteristics of matter; they do not tell what it really is. In the present state of knowledge it is, however, impossible to define matter in any other way.

No experiment has yet shown that matter can be created or destroyed by man. It can be changed from one form to another, it can be given certain physical and certain chemical characteristics, more or less at will, but the actual quantity of matter concerned is always the same after and before such changes. It is customary to state this experience in the form of a law known as the **Law of the Conservation of Matter,** which states that the "*total quantity of matter in the universe is constant.*"

Matter is known to exist in several physical states or conditions of aggregation. The three most familiar are (1) *solid,* (2) *liquid* and (3) *gaseous.* In each of these states matter is conceived as made up of minute particles called molecules which in turn are apparently composed of still smaller parts known as atoms. These atoms can also be broken into parts, but for the purposes of this book it is not necessary to consider such further subdivision.

Experiment and mathematical reasoning seem to indicate that the molecules of all materials are in constant motion and that there are neutralizing attractive and repulsive forces acting between them. In solids the molecules are apparently bound together in such a way that, although they are in constant motion, the external form or shape of the body tends to remain constant; in fact it requires the expenditure of force to cause a change of form. In liquids the molecular attraction is so altered that practically all rigidity disappears and the shape assumed by the liquid is determined by that of the surrounding surfaces, as, for instance, the shape of the vessel containing the liquid. In gases the molecules are still more free and actually tend to move apart as far as possible, so that a gas will spread in all directions until it fills any closed containing vessel.

2. Energy. Nearly everyone has a conception of what is meant by the term energy, but no one yet knows what energy really is. It is defined as *the capacity for doing work*, or *the ability to overcome resistance*. A man is said to be very energetic or to be possessed of a great deal of energy when he has the ability to perform a great amount of work or to overcome great resistances. Matter is said to be possessed of energy when it can perform work or overcome resistance. Actually, matter is not known in any form in which it is not possessed of energy.

There are many different forms of energy. A body in motion can do work and is said to be possessed of **mechanical energy.** A body which we recognize as hot can do work at the expense of the heat associated with it and is said to be possessed of **heat energy.** Light, sound and electricity are all forms of energy.

Experiment and experience have never shown that energy can be destroyed or created by man, but they have shown that one form of energy can be converted into another form under proper conditions. The first part of this experience is stated as a law known as the **Law of the Conservation of**

Energy. This law states that *"the total quantity of energy in the universe is constant."*

3. Units of Matter and of Energy. When attempts are made to measure the amount of anything, some unit of measurement is adopted. Matter is measured in numerous ways and many units are used. The common methods of measuring matter are by volume and by weight. Engineers in English-speaking countries use the cubic yard, the cubic foot or the cubic inch as units in measuring matter by volume and they use the pound, the ounce, the grain, etc. as units in measuring matter by weight.

Energy is measured in many units and, in general, there is a characteristic unit or set of units for each form in which it occurs. Thus the **foot-pound** is very commonly used for measuring *mechanical energy;* the **British thermal unit** for measuring *heat energy;* and the **joule** for measuring *electrical energy.* Some of these units will be defined and considered in greater detail in subsequent paragraphs.

4. Work. Work is defined as *the overcoming of a resistance through a distance.* Thus, work is done when a weight is raised against the resistance offered by gravity; work is done when a spring is compressed against the resistance which the metal offers to change of shape; work is done when a body is moved over another against the resistance offered by friction.

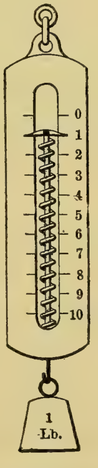


FIG. 1.

The unit of work is the quantity of work which must be done in *raising a weight of one pound through a vertical distance of one foot.* It is called the **foot-pound.** Thus, one foot-pound of work must be done in raising one pound one foot; two foot-pounds of work must be done in raising two pounds one foot or in raising one pound two feet.

If a weight of one pound were suspended from a spring balance as shown in Fig. 1, the balance would indicate a pull or *force* of one pound. No work would be

done by this force as long as the weight remained stationary, because no resistance would be overcome through a distance. If, however, the same weight were slowly or rapidly raised a vertical distance of a foot, one foot-pound of work would be done. A force or pull of one pound would then have overcome a resistance of one pound through a distance of one foot. In general:

$$\begin{aligned} \textit{Work in ft.-lbs.} &= \textit{Resistance overcome in lbs.} \times \textit{distance.} \\ &= \textit{Force in lbs.} \times \textit{distance in ft.} \end{aligned}$$

so that if a force of 10 lbs. pushes or pulls anything which offers a resistance of 10 lbs. while that something travels a distance of, say, 5 ft., the work done will be given by the expression,

$$\begin{aligned} \text{Work} &= 10 \times 5, \\ &= 50 \text{ ft.-lbs.} \end{aligned}$$

A body in falling a certain distance can do work equal to its weight multiplied by the distance it falls because it could theoretically raise an equal weight an equal distance against the action of gravity, and the work done upon this second body would be equal to its weight multiplied by the distance through which it was raised.

It is very important to note that *no work is done by a force if there is no motion;* resistance must be overcome through a distance in order that work may be done. Thus, a force of 1000 lbs. might be required to hold something in position, that is to balance a resistance, but no work would be done if the body upon which the 1000-pound force acted did not move. Again, a weight of 50 lbs. held at a distance of 10 ft. above the surface of the earth would exert a downward push or pull equal to 50 lbs. on whatever held it in that position; it would, however, do no work if held in that position. If allowed to fall through the distance of 10 ft. it could do $50 \times 10 = 500$ ft.-lbs. of work.

It is very convenient to represent graphically the action

of forces overcoming resistances, that is, doing work. This is done by plotting points showing the magnitude of the force at the time that the body on which it is acting has traveled different distances. Thus, suppose a constant force of 10 lbs. pushes a body a distance of 15 ft. against a constant resistance of 10 lbs. The force acting on the body will have a value of 10 lbs. just as the body starts to move, a value of 10 lbs. when the body has moved 1 ft., a value of 10 lbs. when the body has moved 2 ft., and so on. This might be represented by points on squared paper as shown

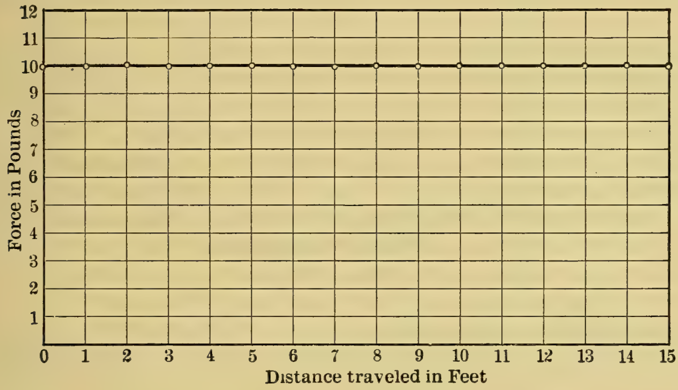


Fig. 2.

in Fig. 2 or by a horizontal line joining those points as shown in the same figure.

The work done by this force would be $10 \times 15 = 150$ ft.-lbs. according to our previous definition. But 10×15 is also the number of small squares under the line representing the action of this force in Fig. 2. The number of these small squares then must be a measure of the work done, but it is also a measure of the area under the line representing the action of the force, so that this area must be a measure of the work done. Each small square represents 1 lb. by its vertical dimension and 1 ft. by its horizontal dimension,

so that its area must represent 1 lb.$\times$1 ft.$=$1 ft.-lb. The total number of squares below the line equals $10\times 15=150$, and since the area of each one represents 1 ft.-lb. the total area under the line represents $150\times 1=150$ ft.-lbs.

It is not always convenient to choose such simple scales as those just used. Thus it might be more convenient to plot the action of this force as is done in Fig. 3. Here the height of a square represents 2 lbs. and the width represents 1 ft.; the area then represents $2\times 1=2$ ft.-lbs. There are $5\times 15=75$ squares under the line and as each

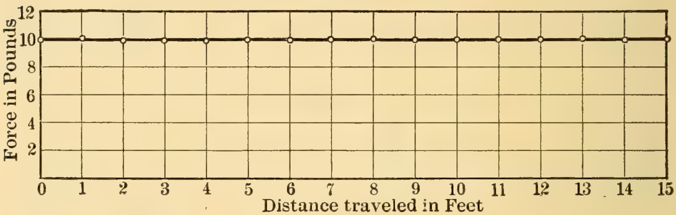


FIG. 3.

represents 2 ft.-lbs. the total area under the line represents $2\times 75=150$ ft.-lbs. as before.

This is a very useful property of these diagrams and the area under the line representing the action of the force always represents the work done, no matter what the shape of that line.

Thus, assume a force which compresses a spring a distance of 6 ins. Suppose that a force of 10 lbs. is required to compress the spring 1 in., a force of 20 lbs. to compress it 2 ins., and so on up to a force of 60 lbs. to compress it 6 ins. Starting with a force of zero, the force will have to gradually increase as the spring is compressed, as shown by the line in Fig. 4. The area of each of the small squares will represent $10\times\frac{1}{12}=\frac{10}{12}$ ft.-lbs. Under the line there is an area equal

to $\frac{6\times6}{2}=18$ small squares, and the work done in compressing the spring must then be $18\times\frac{10}{12}=15$ ft.-lbs.

5. Mechanical Energy. Any body which exists in such a position or location that it could do work by dropping or falling is said to be possessed of **potential mechanical energy,** or of mechanical energy *due to position.* As long as it remains in this position, it cannot do work at the expense of this energy, but, if allowed to fall, it could do so. The

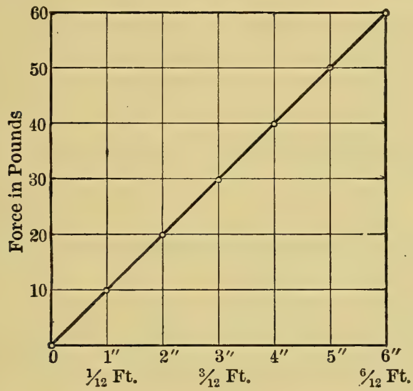


FIG. 4.—Graph Showing Action of Spring.

work it could do would be equal to the product of its weight by the distance it could fall and the potential energy it possesses before starting to fall is measured by this work. Thus, a body weighing 40 lbs. located 10 ft. above the surface of the earth could do $40\times10=400$ ft.-lbs. of work in falling, and, therefore, it is said to be possessed of 400 ft.-lbs. of potential energy before it starts to fall.

If in falling it raises a weight equal to its own (theoretically) through a distance equal to that through which it falls (theoretically), it will have used up 400 ft.-lbs. of energy in doing 400 ft.-lbs. of work upon the body raised

and will no longer be possessed of that amount of potential energy. The body which has been raised will, however, have an equal amount of energy stored in it and will in turn be able to do 400 ft.-lbs. of work if allowed to fall a distance of 10 ft.

If the body assumed above falls through a distance of 10 ft. without raising another body or doing an equivalent amount of work in some other way, it acquires a high velocity. When it arrives at the bottom of the fall of 10 ft., it certainly does not possess the 400 ft.-lbs. of potential energy which it had before dropping nor has it done work at the expense of that energy. Moreover, the energy could not have been destroyed because it is indestructible. The only conclusion is that it must still be possessed of this energy in some way. At the end of the fall it has lost its advantageous position, but it has acquired a high velocity, and experience shows that if brought to rest it can do work upon that which brings it to rest equal to what it could have done in raising a weight as previously described.

At the end of its fall and before being brought to rest, the body is therefore said to be possessed of energy by virtue of its velocity, and this form of energy is called **kinetic mechanical energy.** The *kinetic energy will be exactly equal to the potential energy which disappeared as the body fell.*

Any body which is moving is possessed of kinetic energy because it can do work on anything which brings it to rest. This energy is expressed by the equation,

$$\textit{Kinetic Energy in ft.-lbs.} = \frac{1}{2} \times \frac{W}{32.2} \times V^2,$$

in which

W = the weight of the moving body in pounds.

V = the velocity in ft. per second, and

32.2 = a gravitational constant commonly represented by g.

6. Heat. One of the most familiar forms of energy is *heat,* which manifests itself to man through the sense of touch. In reality every body with which man is familiar possesses an unknown amount of heat energy and it is assumed that this heat energy is in some way associated with the motions and relative positions of the molecules and their constituents.

For this reason heat is often described as **molecular activity** and is regarded as *energy stored up in a substance by virtue of its molecular condition.* Heat energy can be made to perform work in ways which will be discussed later and this is proof that it is a form of energy and not a material substance, as was once supposed.

Heat is observed and recorded by its effects on matter, producing changes in the dimensions or volumes of objects; changes of internal stress; changes of state, as ice to water and water to steam; changes of temperature; and electrical and chemical effects.

Neglecting certain atomic phenomena not yet well understood, the probable source of all heat energy appearing on the earth is the sun. Heat, however, may be obtained from mechanical and electrical energy; from chemical changes; from changes of physical state; from the internal heat of the earth.

7. Temperature. Man early realized that under certain conditions bodies felt " hotter " than under other conditions and gradually came to speak of the " degree of hotness " as the **temperature** of the body. It was later realized that what was really measured as the " hotness " or intensity of heat or temperature of a body was *the ability of that body to transmit heat to others* and that it had no connection with quantity of heat.

Thus, if the temperature of two adjacent bodies happened to be the same, one of them could not lose heat by transmitting it to the other, but if the temperature of one happened to be higher than that of another, the body at

higher temperature would always lose heat to the one at lower temperature.

As a means of *measuring temperature* certain arbitrary scales have been chosen. The **centigrade scale** of temperature, for instance, is based upon the temperatures of melting ice and boiling water under atmospheric pressure. The temperature difference between boiling water at atmospheric pressure and melting ice at atmospheric pressure is arbitrarily called one hundred degrees of temperature, and the temperature of the melting ice is called *zero,* making that of the boiling water *100 degrees.*

Any body which has such a temperature that it will not give heat to, or take heat from, melting ice is said to be at a temperature of zero degrees centigrade, represented as 0° C. Similarly, any body in such a condition that it will not give heat to or take heat from water boiling under atmospheric pressure is said to have a temperature of 100° C. A body with a temperature exactly half way between these two limits would then be said to have a temperature of 50° C.

8. Measurement of Temperature. The temperatures of bodies could be determined by bringing them in contact with such things as melting ice and boiling water and determining whether or not a transfer of heat occurred, but this would be a very cumbersome and unsatisfactory method. As a consequence many other means have been devised for the measurement of temperature.

One of the most common and convenient methods involves the use of what are known as **mercury thermometers.** These depend upon the fact that the expansion of mercury with changing temperature is very uniform over a wide temperature range. Thus, if mercury expands a certain amount when its temperature is raised from that of melting ice to that of boiling water, i.e., 100° C., it will expand just half as much when its temperature is raised half as high, and one-quarter as much when its temperature is raised one-quarter of the range from 0° to 100° C.

The thermometer is made by enclosing a small quantity of mercury in a glass tube fitted with a bulb at one end, as shown in Fig. 5. The lower end of the thermometer is immersed in melting ice and the point on the stem which is reached by the top of the mercury column is marked and labelled 0° C. The thermometer is then immersed in the steam from water boiling under atmospheric pressure and the point reached by the top of the mercury column is marked and labelled 100° C. The distance between the two marks is then divided into one hundred parts and each represents the distance which the end of the column of mercury will move when its temperature changes one centigrade degree.

FIG. 5.—Mercury Thermometer.

It is customary to extend this same scale below 0° and above 100°, carrying it, on expensive thermometers, as far in each direction as the approximation to a constant expansion on the part of the mercury and to constant properties of the glass justifies.

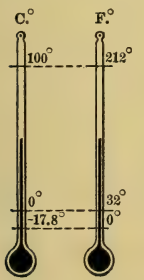


FIG. 6.—Comparison of Centigrade and Fahrenheit Scales.

The temperature of a body can then be found by placing the thermometer in or in contact with that body and noting the point reached by the end of the mercury column. The division reached gives the temperature directly.

The centigrade scale just described is the one commonly used by scientists the world over, but engineers in this country more often use what is known as the **Fahrenheit scale.** This is so chosen that the temperature of melting ice is called 32° F. and the temperature of water boiling under atmospheric pressure is called 212° F. There are

thus 180° on this scale for the same temperature difference as is represented by 100° on the centigrade scale. The relation between the two scales is shown diagrammatically in Fig. 6. It is apparent that the temperature of a body at 0° C. will be 32° F. and that of a body at 0° F. will be −17.8° C.

Since 100 centigrade degrees are equal to 180 Fahrenheit degrees, it follows that

$$1^\circ \text{ C} = \frac{180}{100} = \frac{9}{5}{}^\circ \text{ F.} \qquad (1)$$

and that

$$1^\circ \text{ F} = \frac{100}{180} = \frac{5}{9}{}^\circ \text{ C.} \qquad (2)$$

Therefore, if t_F and t_C represent temperatures on the Fahrenheit and centigrade scales respectively,

$$t_F = \frac{9}{5}t_C + 32. \qquad (3)$$

and

$$t_C = \frac{5}{9}(t_F - 32) \qquad (4)$$

There is still another temperature scale of great importance. It is known as the **absolute scale** and temperatures measured on it are spoken of as *absolute temperatures.* The zero on this scale is located at −273° C. or 273 centigrade degrees below centigrade zero, or, what is the same thing, at −459.4° F., or 459.4 Fahrenheit degrees below Fahrenheit zero. The degrees used are either centigrade or Fahrenheit, as convenient, so that there are absolute temperatures expressed in centigrade degrees above absolute zero and there are absolute tempera-

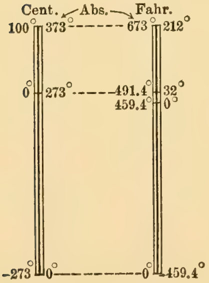


Fig. 7.—Comparison of Absolute and Ordinary Temperature Scales.

tures expressed in Fahrenheit degrees above absolute zero. The relations between the various scales are shown diagrammatically in Fig. 7.

It is apparent from this diagram that,

$$T_F = t_F + 460 \text{ (approximately)} \quad . \quad . \quad . \quad (5)$$

and that

$$T_C = t_C + 273 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

if T_F and T_C represent absolute temperatures and if the number 459.4 is rounded out to 460, as is commonly done.

9. The Unit of Heat Energy. The unit used in the measurement of heat energy in the United States is the **British Thermal Unit** (abbreviated B.t.u). It is defined as *the quantity of heat required to raise the temperature of one pound of pure water one degree Fahrenheit.* In order to make the definition very exact it is necessary to state the temperature of the water before the temperature rise occurs, because it requires different amounts of heat to raise the temperature of a pound of water one degree from different initial temperatures. For ordinary engineering purposes, however, such refinements generally may be omitted.

Many experimenters have shown that *heat energy and mechanical energy are mutually convertible,* that is, the one can be changed into the other. When such a change occurs no energy can be lost since energy is indestructible, and it follows that, if one form is changed into the other, there must be just as much energy present after the change as there was before.

As the units used in measuring the two forms of energy are very different and as it is often necessary to express quantities of energy taking part in such conversions, it is desirable to determine the relations between these units. This was first accurately done by Joule, who showed that one British thermal unit of heat energy resulted from the con-

version of 772 ft.-lbs. of mechanical energy. Later experimenters have shown that the number 778 more nearly expresses the truth than does the number 772 and the larger value is now known as **Joule's Equivalent.**

Expressed mathematically, the relation between the units is

$$1 \text{ B.t.u.} = 778 \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$1 \text{ ft.-lb.} = \frac{1}{778} \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

10. Specific Heat. The specific heat of a substance is defined as *that quantity of heat which is used up or recovered when the temperature of one pound of the material in question is raised or lowered one degree.* Its numerical value depends upon the specific heat of water since the quantity of heat is measured in units dependent upon the amount required to raise the temperature of water. The specific heat of water is, however, very variable, as shown by the values given in Table I., and it is therefore evident that exact numerical values of specific heats can only be given when the definition of the B.t.u. is exactly expressed.

The specific heats of all real substances vary with temperature and the values commonly used are either rough averages or are those determined by experiments at one temperature. For most engineering purposes errors arising from this source may, however, be neglected.

From the definition of specific heat it follows that:

$$C = \frac{Q}{W(t_2 - t_1)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

in which

C = a mean or average specific heat over a range of temperature from t_1 to t_2, and

Q = the heat supplied to raise the temperature of W pounds of material from t_1 to t_2.

TABLE I

Specific Heats of Water.*

(Value at 55° F. taken as unity)

Temp. F°.	Spec. Ht.	Temp. F°.	Spec. Ht.
20	1.0168	350	1.045
30	1.0098	400	1.064
40	1.0045	450	1.086
50	1.0012	500	1.112
60	0.9990	510	1.117
70	0.9977	520	1.123
80	0.9970	530	1.128
90	0.9967	540	1.134
100	0.9967	550	1.140
120	0.9974	560	1.146
140	0.9986	570	1.152
160	1.0002	580	1.158
180	1.0019	590	1.165
200	1.0039	600	1.172
220	1.007		
240	1.012		
260	1.018		
280	1.023		
300	1.029		

* Values taken from Marks and Davis, "Steam Tables and Diagrams," p. 68.

ILLUSTRATIVE PROBLEMS

1. Given: Sp. ht. of iron $=0.113$, of aluminum $=0.211$; Initial temp. $=150°$ F. Temp. range $(t_2-t_1)=100°$ F.

If 1 lb. of iron and 1 lb. of aluminum are cooled through this temperature range, how much more heat is lost in one case than in the other?

$$Q_{al}=WC_{al}(t_2-t_1)=1\times .211\times 100=21.1 \text{ B.t.u.}$$

$$Q_{ir}=WC_{ir}(t_2-t_1)=1\times .113\times 100=11.3 \text{ B.t.u.}$$

$$\text{Difference } 9.8 \text{ B.t.u.}$$

2. If the difference obtained in Prob. 1 were used to heat up 5 lbs. of silver, with a specific heat equal to 0.057, what would be the temperature range through which it would be raised?

$$Q=9.8=5\times 0.057(t_2-t_1)=0.285(t_2-t_1)$$

$$\therefore\quad t_2-t_1=34.4° \text{ F.}$$

3. If the initial temperature of the silver in Prob. 2 were 150° F. what would be the final *absolute* temperature Fahr.?

$$t_2 = t_1 + 34.4° = 150 + 34.4 = 184° \text{ (approximately)}.$$

$$T_2 = 460 + 184 = 644° \text{ F. Abs.}$$

4. 100 lbs. of water in a 20-lb. tank of iron, both at 60° F., are placed in salt brine at 0° F. The water becomes ice at 32° F. and the temperature of the ice is lowered to 26° F., the brine being raised to 26° F. Sp. ht. water = 1.0; Sp. ht. ice = 0.5; Sp. ht. iron = 0.113; Sp. ht. brine = 0.8; and 143 B.t.u. per pound of water must be removed to convert liquid water at 32° F. to ice at the same temperature. What weight of brine is required?

$$100[1(60-32)+143+.5(32-26)]+20\times0.113(60-26)$$
$$= W\times0.8(26-0)$$

$$W = 840 \text{ lbs. of brine.}$$

11. Quantity of Heat. It is impossible to determine the total quantity of heat in or " associated with " a substance, because no means of removing and measuring all the heat contained in any real material have ever been devised. Since, however, the engineer is concerned with changes of heat content rather than with the total amount of heat contained, this fact causes him no difficulty.

For convenience in figuring changes of heat content, it is customary to *assume some arbitrary starting point or datum* and to call the heat in the material in question zero at that point.

Thus, for example, if it were necessary to figure heat changes experienced by a piece of iron weighing 5 lbs. and having a specific heat of 0.1138, and the temperature of this iron never dropped below 40° F. under the conditions existing, this temperature might be taken as an arbitrary starting point above which to figure heat contents. If the iron were later found at a temperature of 75° F., " the heat content above 40° F." would be said to be

$$Q = CW\ (t_2 - t_1) = 0.1138\times5(75-40) = 2.27 \text{ B.t.u.}$$

This type of formula can only be used when the substance does not change its state between the limits of temperature concerned. In the case of water which might change to steam during such a rise of temperature, it might be necessary to include other heat quantities in the calculations, as shown in a later chapter.

12. Work and Power. Since steam engines are designed for the purpose of converting the heat energy contained in fuel into mechanical energy which may be used to perform work, it will be necessary to consider the units used in measuring work and power.

Work was defined in a previous paragraph as *the overcoming of a resistance through a distance*, by the application of a force; that is, *a force expressed in pounds, multiplied by the distance in feet through which the force acts, gives a product expressed in foot-pounds.*

The amount of work performed in a *unit of time* is termed **power,** which may be defined as the rate of doing work. Therefore,

$$\text{Power} = \frac{\text{Force} \times \text{Distance}}{\text{Time (min. or sec.)}}. \quad . \quad . \quad . \quad (10)$$

The unit of power used by steam engineers is the **horsepower,** which is equivalent to the performance of *33,000 ft.-lbs. of work per minute,* or 550 ft.-lbs. of work per second, or 1,980,000 ft.-lbs. per hour. Therefore, the horse-power developed by any mechanism is

$$\text{h.p.} = \frac{\text{ft.-lbs. of work per min.}}{33{,}000}. \quad . \quad . \quad . \quad (11)$$

Since 33,000 ft.-lbs. of work can be accomplished only by the expenditure of 33,000 ft.-lbs. of energy and since one B.t.u. of energy is equal to 778 ft.-lbs., it follows that 33,000 ft.-lbs. of work must be the equivalent of $\frac{33{,}000}{778} = 42.41$ B.t.u.

It is customary to speak of power in terms of **horse-**

power-hours. One horse-power-hour means the doing of work equivalent to one horse-power for the period of one hour, or the doing of work at the rate of *33,000 ft.-lbs. per minute for an hour.* A horse-power-hour is therefore equivalent to $33{,}000 \times 60 = 1{,}980{,}000$ ft.-lbs. As 33,000 ft.-lbs. are equivalent to 42.41 B.t.u., it follows that $42.41 \times 60 = 2544.6$ or about 2545 B.t.u. are the equivalent of one horse-power-hour.

The number 2545 should be memorized as it is very often used in steam-power calculations. If an engine could deliver one horse-power-hour for every 2545 B.t.u. it received, it would be working without losses of any kind; that is, all the heat energy entering it would leave it in the form of useful mechanical energy. It will be shown later that this is impossible even in the most perfect or ideal engine.

REVIEW PROBLEMS

1. Express 32° F. in degrees centigrade.

2. Express 150° F. in degrees centigrade.

3. Express 250° C. in degrees Fahrenheit.

4. Express the results of problems 1, 2 and 3 in absolute values.

5. What is the heat equivalent of 233,400 ft.-lbs. of work?

6. Find the heat supplied 10 lbs. of water when its temperature is raised from 20° F. to 160° F., assuming the mean specific heat over this range to be 0.997.

7. Find the temperature change of 2 lbs. of lead (sp. ht. 0.0314) when 20 B.t.u. are added.

8. How many B.t.u. must be abstracted to lower the temperature of 15 lbs. of water from 212° F. to 32° F., assuming the specific heat of water to be unity?

9. Find the weight of water which will have its temperature tripled in value by the addition of 250 B.t.u., the final temperature being 150° F. Assume specific heat unity.

10. The specific heat of a piece of wrought iron is 0.113 and of a given weight of water is 1.015. 1 cu. ft. of water weighs approximately 62.5 lbs. Find the increase in temperature of 4 cu. ft. of water when a common temperature of 65° F. results from placing in the water a piece of iron weighing 15 lbs. at a temperature of 900° F.

11. Find the final temperature of the mixture, when 100 lbs. of iron (sp. ht. =0.113), at a temperature of 1200° F. are immersed in 300 lbs. of water (sp. ht. 1.001) at a temperature of 50° F.

12. Five pounds of silver (sp. ht. =0.057) at 800° F. are immersed in water at 60° F., resulting in a final temperature of 85° F. Assume Sp. ht. water = 1. What weight of water is necessary?

13. An engine is developing 10 horse-power. Express this in ft.-lbs. of work done per minute and find the amount of heat energy equivalent to this quantity of mechanical energy.

14. A pump raises 1000 lbs. of water 50 ft. every minute. How much work is done? Find the equivalent horse-power.

15. An engine develops 1,980,000 ft.-lbs. of work at the fly-wheel per minute.

(*a*) Find the horse-power developed.

(*b*) If this engine operated in this way for an hour, how many horse-power hours would it make available?

(*c*) What would be the equivalent of this number of horse-power hours in British thermal units?

CHAPTER II

THE HEAT-POWER PLANT

13. The Simple Steam-Power Plant. The various pieces of apparatus necessary for the proper conversion of heat energy into mechanical power constitute what may be called a "Heat-Power Plant," just as the apparatus used in obtaining mechanical energy from moving water is called an hydraulic or water-power plant. Heat-power plants are distinguished as "Steam-Power Plants"; "Gas-Power Plants"; etc., according to the way in which the heat of the fuel happens to be utilized.

The apparatus around which the plant as a whole centers, that is, the apparatus in which heat energy is received and from which mechanical energy is delivered, is termed the **engine** or **prime-mover.** This heat engine may use *steam* generated in boilers and may require certain apparatus, such as condensers, pumps, etc., for proper operation; or it may use *gas*, generated in gas-producers requiring coolers, scrubbers, tar extractors and holders, depending upon the class of fuel used and upon certain commercial considerations. Again, the power-plant may simply contain an internal-combustion engine using natural gas, gasoline or oil, a type of plant which is now very common.

But whatever type of plant is used, a general method of operation is common to all. *Heat energy in fuel is constantly fed in at one end of the system and mechanical energy is delivered at the other end.* The steam-power plant will be briefly described in the following paragraphs, showing the cycle of events with the attendant losses through the system.

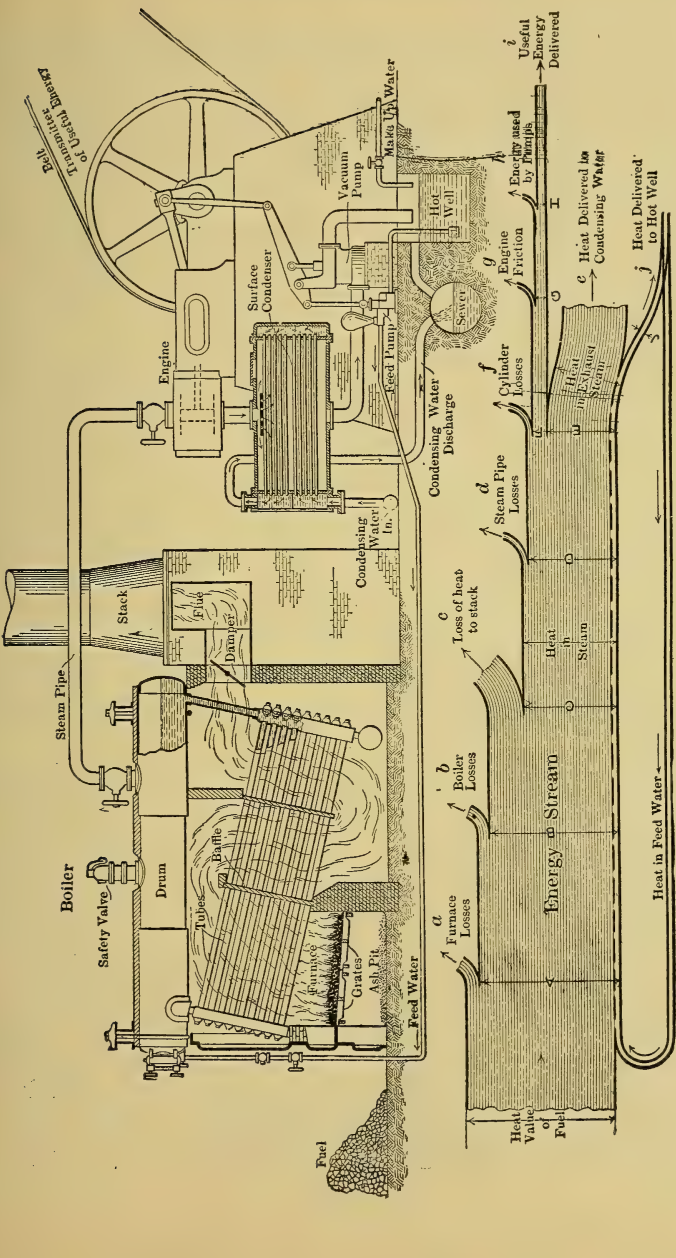


FIG. 8.—Elements of a Steam Power Plant.

In Fig. 8 is shown a simple steam-power plant which converts into mechanical energy part of the heat energy, originally stored in coal, by means of a prime-mover called a *steam-engine*. The main pieces of apparatus used in this type of plant are the *steam-boiler;* the *steam-engine;* the *condenser;* the *vacuum pump;* and the *feed-pump*. The energy stream shows the various losses occurring throughout the plant. These losses cause the "delivered energy" stream to be only a small fraction of the total heat sent into the system.

14. Cycle of Events. 1. Fuel is charged on the grate under the boiler, where it is burned with the liberation of a large amount of energy. Air is drawn or forced through the grates in proper proportions to support this combustion. The hot gases resulting pass over the tubes, in a definite path set by the baffle plates, so that the largest possible amount of heating surface may be presented to the products of combustion.

There are certain losses accompanying this operation, such as radiation, loss of volatile fuel passing off unburned, loss of fuel through the grate, and loss of heat through the excess air which must always be supplied to insure combustion.

2. That part of the heat in the gases which is not lost by radiation from the boiler and in the hot gases flowing up the stack passes through the heating surfaces of the boiler to the water within. From 50 to 80 per cent of the total heat energy in the fuel passes through the heating surfaces and serves to raise the temperature of the water to the boiling point at the pressure maintained, and to convert this water into steam according to the requirements.

3. Having obtained steam within the boiler, it is led through a system of pipes to a steam engine, where some of the heat stored in the steam is converted into mechanical energy by the action of that steam against a piston. The steam is then discharged, or exhausted, from the engine

at a much lower temperature and pressure than when it entered.

From 5 to 22 per cent of the available heat in the steam is converted into mechanical energy in the engine cylinder, and because of frictional and other losses occurring in the mechanism, only from 85 to 95 per cent of this energy is turned into useful work at the fly-wheel.

4. In some plants, known as **non-condensing plants,** the exhaust steam, which still contains the greater part of all the heat received in the boiler, is discharged to the atmosphere and represents a complete loss. In others, known as **condensing plants,** the exhaust steam is led to a condenser, where it is condensed by cold water, which absorbs and carries away the greater quantity of the heat not utilized in the engine. The condensed steam or " condensate " is then either discharged to the sewer or transferred by means of a *vacuum-pump* to the *hot-well,* from which it is drawn by means of the *feed-water-pump,* raised to the original pressure of the steam, and returned to the boiler. Here it is again turned into steam and the cycle of operations outlined above is repeated. Naturally there is some loss due to evaporation and leaks throughout the system, so that " make-up " water must be supplied.

The series of events just described constitutes a complete, closed **cycle** of operations, wherein the water is heated, vaporized, condensed and returned to the boiler, having served only as a medium for the transfer of heat energy from fuel to engine and the conversion of part of that energy within the cylinder. The water in such a case is known as the *working substance.*

It is often more convenient to discard the working substance after it leaves the cylinder, as suggested above in the case of a non-condensing plant; or, as in the case of a gas engine, where a new supply of working substance must be supplied for each cycle, because the burned gases of the previous cycle cannot be used again.

15. Action of Steam in the Cylinder. In order to prepare for the more detailed discussion of the action of the steam in the engine cylinder, to be taken up in a later chapter, a brief outline of the events occurring within the prime-mover will be considered at this point.

Steam enters the cylinder through some kind of an *admission valve,* and acts upon the piston, just as the latter has approximately reached one end of its stroke and is ready to return. The heat-energy stored up in the steam causes it to expand behind the piston, thereby driving the latter out and performing work at the fly-wheel. At about 90 or 95 per cent of the stroke, the exhaust valve opens, and the steam begins to exhaust, the pressure within the cylinder dropping almost to atmospheric or to that maintained in the condenser by the time the piston has reached the end of its stroke. . On the next or return stroke the remaining steam is forced out through the exhaust port, until, at some point before the end of the piston travel, the exhaust valve closes, and the low-pressure steam trapped in the cylinder is compressed into the clearance space so that its pressure rises. Admission then occurs, and the cycle is repeated.

The diagram given in Fig. 9 illustrates the operation of steam within the cylinder. This diagram is plotted between pressures of steam within the cylinder as ordinates and corresponding piston positions as abscissas.

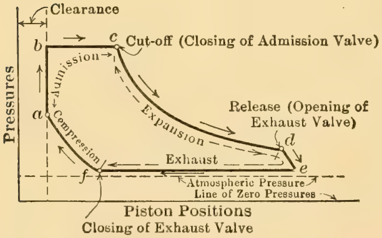


FIG. 9.—Steam Engine Indicator Diagram.

The method of obtaining such a diagram, known as an **indicator-diagram,** will be fully described in a later chapter. Since vertical ordinates represent pressure in pounds per square inch, and horizontal abscissas repre-

sent feet moved through by the piston, the product of these two must be work. But the product of vertical by horizontal distances must also give area. Therefore, by

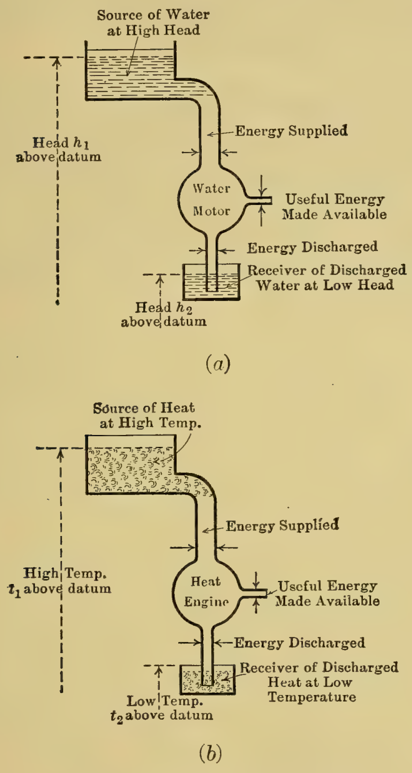


FIG. 10.—Hydraulic Analogy.

finding the area enclosed within the bounding lines of the cycle and multiplying this by a proper factor, the foot-pounds of work developed within the cylinder can be determined.

16. Hydraulic Analogy. The operation of heat-engines is analogous to that of water-wheels. A water-wheel de-

velops mechanical energy by receiving water under a high head, absorbing some of its energy, and then rejecting the fluid under a low head. Similarly, the heat-engine receives heat energy at a high temperature (head), absorbs some of it by conversion into mechanical energy, and then rejects the rest at a low temperature (head).

The analogy can be carried still further. The water-wheel cannot remove all the energy from the water, nor can the heat-engine remove all the heat-energy from the working substance. There is a certain loss in the material discharged in both cases and this cannot be avoided.

This analogy is illustrated diagrammatically in Fig. 10 (*a*) and (*b*) in which the widths of the streams represent quantity of energy.

CHAPTER III

STEAM

17. Vapors and Gases. When a solid is heated, under the proper pressure conditions, it ultimately melts or fuses and becomes a liquid. The temperature at which this occurs depends upon the particular material in question and upon the pressure under which it exists. Ice, which is merely solid water, melts at 32° F. under atmospheric pressure, while iron melts at about 2000° F. under atmospheric pressure.

When a liquid is heated, it ultimately becomes a gas, similar to the air and other familiar gases. If this gas is heated to a very high temperature and if the pressure under which it is held is not too great, it *very nearly* obeys certain laws which are simple and which are called the *laws of ideal gases*.

When the material is in a state between that of a liquid and that in which it very nearly obeys the laws of ideal gases, it is generally spoken of as a **vapor.** This term is used in several different ways and with several different modifying adjectives which will be explained in greater detail in later sections.

18. Properties of Steam. Of the many vapors used by the engineer, steam or water vapor is probably the most important, because of the ease with which it can be formed and also because of the tremendous field in which it can be used. It is generated in a vessel known as a *steam boiler*, which is constructed of metal in such a way that it can contain water, and that heat energy, liberated from burning fuel, can be passed into the water, converting part or all of it into water vapor, that is, into steam.

The properties of water vapor must be thoroughly understood before the steam engine and steam boiler can be studied profitably. Probably the easiest way of becoming familiar with these properties is to study the use made of heat in the generation of steam from cold water.

19. Generation of Steam or Water Vapor. For the purposes of development, assume a vessel of cylindrical form, fitted with a frictionless piston of known weight, as shown in Fig. 11, (*a*)and (*b*), the whole apparatus being placed under a bell-jar in which a perfect vacuum is maintained.

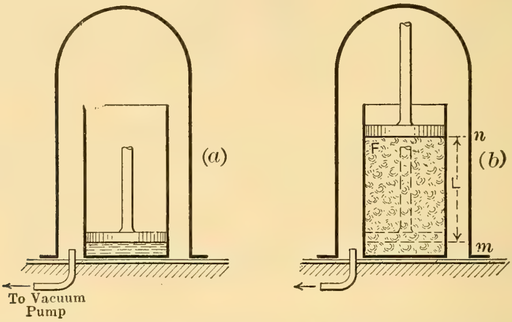


FIG. 11.—Formation of Steam at Constant Pressure.

Assume *one pound* of water in the cylinder, with the piston resting on the surface of the liquid. There will be some definite pressure exerted by the piston upon the surface of the liquid, and its value will be determined entirely by the weight of the piston.

It is convenient in engineering practice to refer all vaporization phenomena to some datum temperature, and since the melting point of ice, 32° F., is a convenient reference point, it is used as a standard datum temperature, in practically all steam-engineering work. Therefore, assuming the water in the jar to be at 32° F., if heat is applied the temperature of the liquid will rise approximately 1° F.

for every B.t.u. of heat added, since the specific heat of water is approximately unity.

Experiment shows that *for each pressure under which the water may exist some definite temperature will be attained at which further rise of temperature will cease and the liquid will*

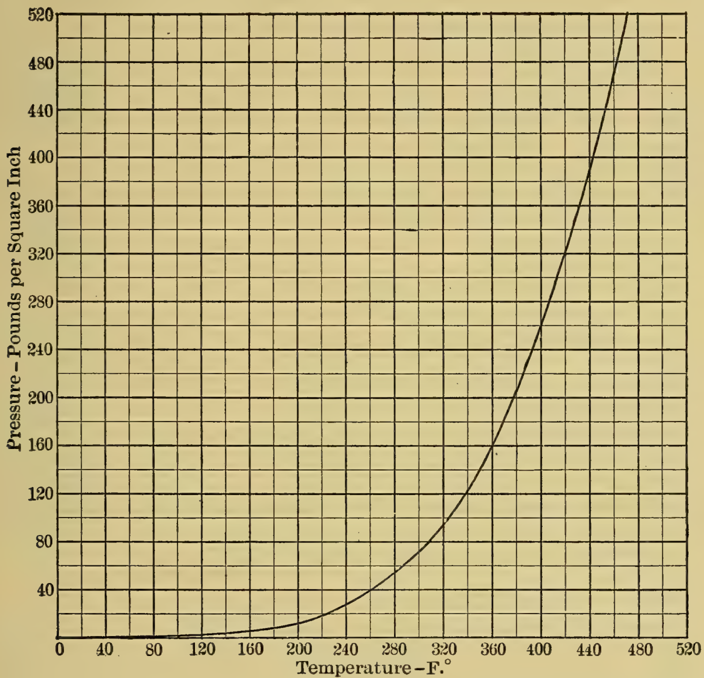


Fig. 12.—Pressure-Temperature Relations, Saturated Water Vapor.

begin to change to a vapor, that is, to vaporize. The temperatures at which vaporization occurs at different pressures are called the **temperatures of vaporization** at those pressures.

The temperatures of vaporization of water are plotted against pressure in Fig. 12. It should be noted that the values of vaporization temperature increase very rapidly for small pressure changes in the case of low pressures, but

that, for the higher pressures, the variation of temperature is very small for enormous variations of pressure. This fact is of great importance in steam engineering.

The temperatures of vaporization are tabulated with other properties of water vapor in so-called **steam tables** and are constantly referred to by engineers. An example of such a table is given on pp. 392 to 399.

Returning now to the apparatus under discussion, as heat is supplied, the temperature of the water will rise from 32° F. until it reaches the temperature of vaporization corresponding to the pressure exerted upon the water by the piston. When this temperature is reached vaporization will begin, and if sufficient heat is supplied, will continue *without change of temperature* until the water is entirely converted into vapor.

Up to the time at which vaporization starts the volume of the water will change very little, so that the piston will be raised only a negligibly small amount and practically no work will be done upon it by the water. On the other hand, when vaporization occurs the volume of the material will change by a very large amount and the piston will be driven out (raised) against the action of gravity. That is, *work will be done by the steam* in driving the piston out during the increase in volume which accompanies vaporization.

It is found that a very great quantity of heat is used up during the process of vaporization despite the fact that no temperature change occurs. This is described by saying that the heat which is supplied during this period becomes *latent,* that is, not apparent, and the quantity of heat is therefore spoken of as the **latent heat of vaporization.** It is assumed to consist of two parts, that used for separating the liquid molecules against their attractive forces and that used for doing the work which is done upon the piston as it is moved upward. The former is called the **internal latent heat** because it is used for doing internal or intermolecular

work; the latter is called **external latent heat** because it is used for the doing of external work.

It is to be noted that the internal latent heat may be assumed to be tied up in some way within the molecular structure of the material and hence to be *in the steam.* The external latent heat, on the other hand, is used up as fast as supplied for the purpose of driving the piston out against the action of gravity. When the piston has been raised to any point, the energy used in raising it is not *in* the steam, but is stored as potential energy *in the piston.* To get it back the piston must be allowed to drop. The term "external" is therefore well chosen; the external latent heat is in no sense *in* the steam; *it is stored in external bodies or mechanism.*

After the constant temperature vaporization is complete, the further addition of heat will again cause a rise of temperature and a gradual increase of volume. Such raising of the temperature of steam already formed is called **superheating** and results in carrying the vapor nearer and nearer to the condition in which it very nearly obeys the laws of ideal gases. Since an increase of volume accompanies superheating, the molecules of the vapor must move farther and farther apart as superheating progresses.

Vapor in the condition in which it is formed from the liquid and which has the same temperature as the liquid from which it was formed is called **saturated vapor.** This term can be pictured as meaning that the maximum number of molecules of vapor are packed into a given space; the addition of heat to saturated vapor would cause superheating and the separation of the molecules so that fewer could be contained in a given space.

20. Heat of Liquid, q or h. Returning once more to the start of the process described in the preceding section, heat was added to water initially at 32° F. until the temperature of vaporization corresponding to the existing pressure was attained. The heat added during this period is

called the **heat of the liquid**, and is usually designated by the letters q or h. If the mean specific heat of water at constant pressure (C_p) for the temperature range under consideration were constant, q would be given by the equation

$$q=C_p(t_v-32)$$

in which t_v is the temperature of vaporization, and if C_p were equal to 1, it would follow that

$$q=t_v-32. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Therefore, if water boils under a pressure of 50 lbs. at a temperature, read from the steam tables, of 281° F., it would follow that for 50 lbs.

$$q=281-32=249 \text{ B.t.u.}$$

But the steam tables (see p. 394) for this pressure (50 lbs.) give $q=250.1$ B.t.u., indicating, as was shown in Chap. I., that the specific heat of water does not remain constant, and for this case the mean value must have been approximately 1.004 as indicated by the following calculation.

$$q=C_p\ (t_v-32) \text{ or } 250.1=C_p\times 249$$

so that

$$C_p=\frac{250.1}{249}=1.004+$$

Hence it is always advisable to use the steam table values of q, except for very approximate calculations.

21. Latent Heat of Vaporization, r or L. The heat supplied during the period of vaporization has already been referred to as the latent heat of vaporization and has been divided into internal and external latent heats.

The internal latent heat is generally designated by ρ or by I and the external latent heat by the group of letters APu or by E. The group APu merely represents the prod-

uct of pressure, P, by volume change during vaporization, u, and by the fraction $\frac{1}{778}$ which is represented by A. The product of the first two terms gives external work in foot-pounds during vaporization, and dividing this by 778 (Joule's Equivalent) converts it to heat units to correspond with the other values. It should be noted that P in this expression stands for pressure in pounds per square foot.

The total latent heat of vaporization is generally designated by r or by L, and it follows from what has preceded that

$$r = \rho + APu. \quad . \quad . \quad . \quad . \quad . \quad (13)$$

The value of r for atmospheric pressure, that is, for a temperature of vaporization of 212° F., is very often used in engineering and should be memorized. Its value is now generally taken as 970.4 B.t.u., though recent work would seem to indicate a value of about 972 as nearer the truth.

22. Total Heat of Dry Saturated Steam, λ or H. The total heat required to convert a pound of water at 32° F. into a pound of saturated vapor at some temperature t_v is called the *total heat of the steam* or the heat above 32° and is designated by λ or by H. It is obviously the sum of the quantities which have just been considered, so that

$$\lambda = q + r = q + \rho + APu. \quad . \quad . \quad . \quad . \quad (14)$$

23. Total Heat of Wet Steam. In practical work the engineer seldom deals with pure saturated steam, the saturated vapor nearly always carrying in suspension more or less liquid water at its own temperature. To distinguish between saturated steam which carries liquid water and that which does not, the former is called **wet steam** or wet saturated steam, and the latter **dry saturated steam.**

The condition of dryness or wetness is described by what is known as the **quality** of the steam. Dry saturated steam is said to have a quality of 100 per cent while saturated steam carrying 10 per cent by weight of liquid is said to

have a quality of 90 per cent. Quality expressed as a decimal fraction is designated by the letter x, so that if x is said to be equal to 0.8 in referring to a certain sample of steam, it means that that steam sample consists of 80 per cent by weight of saturated steam and 20 per cent liquid at the same temperature.

Since the *water in wet steam has the same temperature as the steam*, it contains all the heat of the liquid which it would contain if it had been converted into steam, but it obviously contains no latent heat of vaporization. It follows that the *total heat in a pound of wet steam* (one pound of a mixture of saturated steam and water) with quality equal to x is

$$\text{Heat per pound} = q + xr = q + x\rho + xAPu \quad . \quad . \quad (15)$$

The letter λ should never be used in designating the total heat per pound of wet steam, as it has been chosen as the symbol of the total heat per pound of dry, saturated steam.

24. Heat of Superheat. When the temperature of saturated steam is raised by the addition of more heat, that is, when it is superheated, a very definite quantity of heat is required. The quantity required per pound per degreee would, by definition, be the specific heat of the material in question.

If the specific heat of superheated steam were reasonably constant, the heat required to raise its temperature at constant pressure from saturation temperature to some higher value t_2 would be given by the expression

$$\text{Heat required per pound} = C_p(t_2 - t_s)$$

but superheated steam, as handled by the engineer, is generally comparatively near the saturated condition, and under these circumstances the values of the specific heat vary rapidly with changes of pressure and temperature. The extent of these variations is shown in Fig. 13. It will be observed that for low pressures the specific heat is approxi-

mately constant at a value below 0.5 for any given pressure, but that for very high pressures it varies widely over a comparatively small temperature range. Thus at 600 lbs. per square inch the specific heat changes from unity at about 510° F. to 0.6 at about 550° F.

Practically, it is customary to use the type of equation just given and to substitute a *mean specific heat* over the required temperature range for the specific heat which cannot be assumed constant without too great an error. The

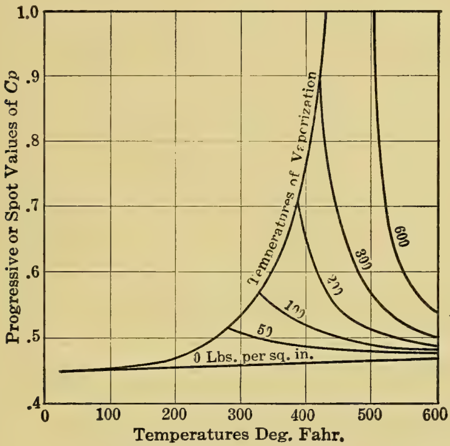


FIG. 13.—Progressive Values of Specific Heat, C_p, Water Vapor.

equation for heat required to raise the temperature from t_v to t_2 is then

$$\text{Heat of superheat per pound} = C_{pm}(t_2 - t_v). \quad . \quad (16)$$

in which C_{pm} stands for the mean specific heat at constant pressure over the temperature range from t_v to t_2.

Values of mean specific heats of superheated steam are given in Fig. 14, the values indicated by the curves giving the mean specific heat between saturation temperatures and various higher temperatures at different pressures.

25. Total Heat of Superheated Steam. The total heat required to convert one pound of water at 32° F. into superheated steam at a temperature of t_2° F. under constant pressure conditions is obviously

$$\text{Total heat per pound} = q + r + C_{pm}(t_2 - t_v). \quad . \quad (17)$$

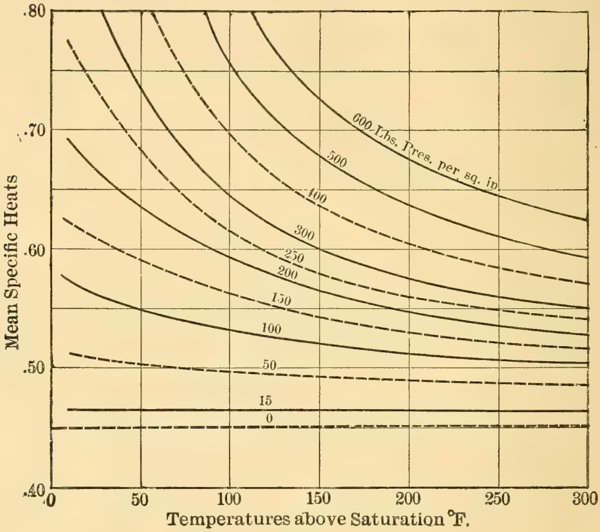


FIG. 14.—Variation of Mean Specific Heat, Water Vapor.

and representing the degrees of superheat $(t_2 - t_v)$ by D, as is customary, this becomes

$$\text{Total heat per pound} = q + r + C_{pm}D. \quad . \quad . \quad (18)$$

26. Specific Volume of Dry Saturated Steam, V or S. The volume occupied by *one pound* of a substance is spoken of as the **specific volume** of that material. In the case of dry saturated steam there are as many specific volumes as there are pressures under which the steam can exist. These values are generally tabulated in steam tables and are represented by the letter V or the letter S.

The values of the specific volumes of steam at different pressures are given in Fig. 15. It is important to note the very gradual change of specific volume at high pressures and the very rapid change and enormous increase at low pressures. These facts have considerable influence on steam engineering practice.

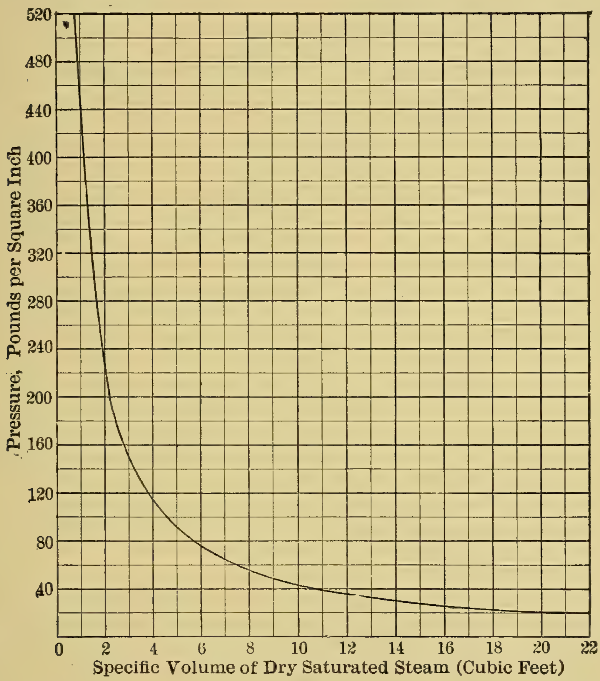


Fig. 15.—Pressure-Volume Relations, Saturated Water Vapor.

A curve giving properties of saturated steam is called a **saturation curve,** so that this name may be, and often is, applied to the curve given in Fig. 15.

The volume occupied at any pressure by half a pound of dry saturated steam will obviously be half that occupied by one pound of such material at the same pressure, and

the same statement can be made for any other fraction of a pound. It follows that if the small volume occupied by liquid water in wet steam be neglected, the volume occupied by one pound of steam (mixture) of 50 per cent quality can be assumed equal to half that occupied by an equal weight of dry saturated steam at the same pressure. A similar statement could of course be made for any other quality and a corresponding fraction.

Hence if one pound of "wet steam" at a given pressure is found to have such a volume that it would be indicated by point b in Fig. 16, the quality of this material must be given by the expression $x = \frac{ab}{ac}$ if the volume occupied by the liquid water in the mixture be neglected.

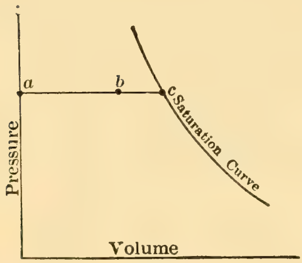


FIG. 16.—Determining Quality from Volume.

27. Specific Density of Dry Saturated Steam, $\frac{1}{V}$ or δ. The *weight per cubic foot* of saturated steam is spoken of as its specific density. The specific density is obviously the reciprocal of the specific volume and is therefore $\frac{1}{V}$.

28. Reversal of the Phenomena Just Described. If any process which has resulted in the absorption of a quantity of heat by a substance be carried through in the reverse direction, the same amount of heat will be liberated. It follows that a pound of dry saturated steam will give up the total latent heat of vaporization when condensed to liquid at the same temperature, and that the resultant pound of hot water will give up the total heat of the liquid if cooled to 32° F.

29. Generation of Steam in Real Steam Boiler. The steam boiler is equivalent to a vessel partly filled with water

and fitted with means for supplying heat to the water and for carrying off the vapor formed. This is shown diagrammatically in Fig. 17. At first glance this would not seem to be at all similar to the cylinder and piston already considered, but it really is the exact equivalent so far as the generation of steam is concerned. The flow of steam out of the steam-pipe is restricted to the extent necessary to maintain a high and constant pressure within the boiler, and, when in regular operation, steam is formed within the

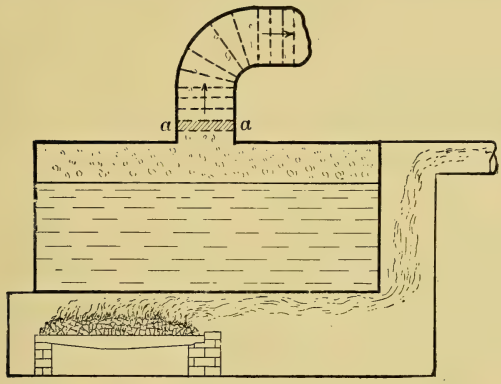


FIG. 17.—Formation of Steam in a Steam Boiler.

boiler under this pressure just as fast as necessary to replace that flowing out.

By picturing the steam as flowing out in layers or lamina these lamina can be imagined as taking the place of the piston in the apparatus of Fig. 11, and each pound of steam formed will then push a piston before it exactly as was assumed in the previous discussion.

30. Gauge Pressure. The steam pressure in a boiler is commonly determined by means of an instrument called a *pressure gauge*. These instruments are almost always constructed about as shown in Fig. 18 (*a*) and (*b*). The Bourdon spring is a tube of elliptical section bent approximately into

the arc of a circle. One end of this tube is connected directly to the pressure connection of the gauge and the other end is closed and connected to a toothed sector as shown.

When the pressure inside a tube of this character is increased, the tube has a tendency to unroll or straighten out, and in so doing it moves the toothed sector in such a way as to rotate the pointer or gauge hand and make its end move over the scale in the direction of increasing pressure. With diminishing pressure the tube again rolls up and rotates the hand in the opposite direction.

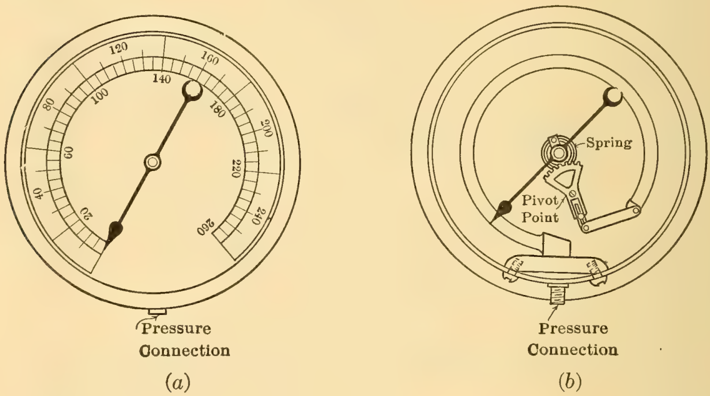


Fig. 18.—Bourdon Pressure Gauge.

Instruments of this kind are so made and adjusted that the hand points to zero when the gauge is left open to the atmosphere. Under such conditions the pressure inside the tube is equal to that of the atmosphere and is not zero. The gauge therefore only indicates pressures above atmospheric on its scale, and the total pressure inside the boiler is really that shown by the gauge plus that of the atmosphere.

Pressures as indicated by the gauge are called **gauge pressures.** Pressures obtained by adding the pressure of

the atmosphere to the reading of the gauge are known as **absolute pressures.** Then

Absolute Pressure = Gauge Pressure + Atmospheric Pressure

and

Gauge Pressure = Absolute Pressure − Atmospheric Pressure.

In accurate work the existing atmospheric pressure should be determined by means of the barometer, but for ordinary, approximate calculations and for cases in which no barometric data are available, it is customary to assume the pressure of the atmosphere to be equal to 14.7 lbs. per square inch. This is very nearly true, on the average, at sea level, but is generally far from true at higher elevations.

PROBLEMS

1. Determine by means of the steam tables the temperatures, total heats, heats of liquid, internal and external latent heats, and the specific volumes of 1 lb. of dry, saturated steam under the following absolute pressures (lbs. per sq. in.): 15, 50, 95, 180 and 400.

2. Determine the heats of the liquid, latent heats of vaporization and total heats for 2 lbs. of dry saturated steam at the following temperatures in °F.: 101.83, 212 and 327.8.

3. Determine the volumes occupied by 2 lbs. of dry saturated steam under the conditions of problem 2.

4. Determine the heats of the liquid, latent heats of vaporization and total heats for 1 lb. of saturated steam with a quality of 90% at the following absolute pressures: 25, 50, 75, 125.

5. Determine the total heat above 32° F. in 12 lbs. of saturated steam with quality of 97% at a pressure of 125 lbs. per square inch absolute.

6. What space will be filled by 20 lbs. of dry saturated steam at a pressure of 150 lbs. per square inch absolute?

7. What space will be filled by 20 lbs. of saturated steam at a pressure of 150 lbs. per square inch absolute and with a quality of 95% if the volume occupied by the water present be neglected?

8. How many pounds of dry saturated steam at a pressure of 75 lbs. per square inch absolute will be required to fill a space of 10 cu. ft.?

9. How many pounds of saturated steam with quality 96% and at a pressure of 110 lbs. per square inch absolute will be required to fill a space of 8 cu. ft.?

10. How much external work, measured in B.t.u., is done when 1 lb. of water at the temperature of 212° F. is converted into dry saturated vapor at the same temperature?

11. How much external work, measured in foot-pounds, is done when 2 lbs. of water at a temperature of 212° F. are converted into 90% quality steam at the same temperature?

12. How much heat is required for doing internal work during the vaporization of 1 lb. of water under such conditions that the total latent heat of vaporization is 852.7 B.t.u. and the external latent heat is 83.3 B.t.u.?

13. What is the quality of steam containing 1000 B.t.u. above 32° F. per pound when under a pressure of 150 lbs. per square inch absolute?

14. Heat is added to 1 lb. of mixed steam and water while the pressure is maintained constant at 100 lbs. per square inch absolute. The percentage of steam in the mixture is increased thereby from 50% to 95%.

(*a*) How much heat was added?

(*b*) How much internal latent heat was added?

(*c*) How much external latent heat was added?

15. How much heat is required to completely vaporize 1000 lbs. of water at a temperature of 92° F. when pumped into a boiler in which steam is generated at a pressure of 150 lbs. per square inch gauge? Note that heat above 32° F. in 92° F. water is given as q in steam tables for a temperature of 92° F.

16. Find the amount of heat necessary to produce in a boiler 200 lbs. of steam having a quality of 97% at a pressure of 100 lbs. gauge when the feed water has a temperature of 205° F.

17. What volume would be occupied by the material leaving the boiler in problem 16, neglecting volume occupied by water?

CHAPTER IV

THE IDEAL STEAM ENGINE

31. The Engine. If the cylinder and piston assumed in the discussion of the last chapter be imagined as turned into a horizontal position and fitted with a frame, piston

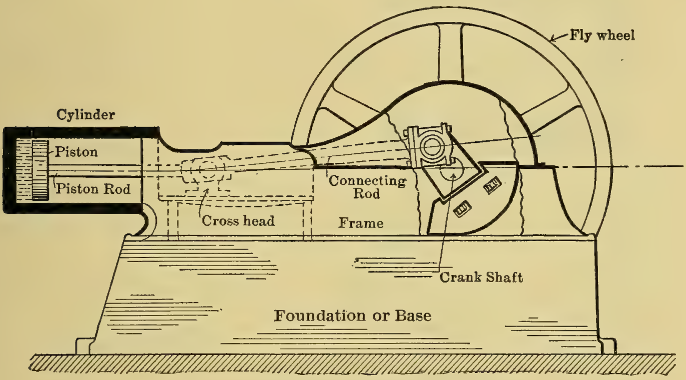


Fig. 19.—Simple Steam Engine.

rod, crosshead, connecting rod, crank shaft and flywheel as in Fig. 19, a device results which might be used as a steam engine for the production of power. By adding heat to, and taking heat from, the water and steam in the cylinder in the proper way and at the proper time, the water and steam, or working substance, can be made to do work upon the piston. The piston can transmit this work through the mechanism to the rim of the flywheel, and it can be taken from the rim by a belt connected to a pulley on a machine which is to be driven.

To make the analysis easier, a simplified type of engine will be assumed. It is shown in Fig. 20 and consists of the same cylinder, piston and piston rod as just described. A wire is fastened to the end of the piston rod and run back over a pulley in such a way that a weight fastened to the free end of the wire will be raised if the piston moves out. The weight is made up of two parts, one large and one small. When both are on the wire the pull which they exert causes the piston to exert a high pressure upon whatever is contained in the cylinder.

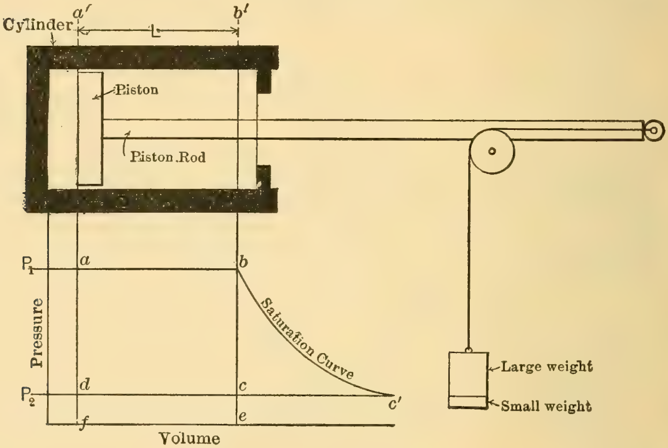


FIG. 20.—Simplified Steam Engine.

When only the small weight hangs on the wire, the piston exerts a much lower pressure upon the material in the cylinder.

Imagine that the piston and the walls of the cylinder are made of some ideal material which will not receive or conduct heat. Imagine also that the cylinder is fitted with a permanent head which is a perfect conductor of heat. These conditions are of course ideal but are assumed for the sake of simplicity.

Assume further that, when one pound of water is con-

tained in the cylinder and the piston is driven into the cylinder by the two weights until the space between the piston and the cylinder head is just large enough to contain the pound of water, the piston exerts a high pressure equal to P_1 pounds per *square foot* against the water. The volume of this water and the pressure upon it can be represented by the point a of the PV diagram, Fig. 20.

32. Operation of the Engine. With conditions as described in the preceding paragraphs, imagine a flame or other source of heat at high temperature to be brought into contact with the conducting cylinder head and to pass heat into the cylinder, raise the temperature of the water within to the temperature of vaporization and ultimately vaporize it. As the water vaporizes it will push the piston out of the cylinder just as described in the last chapter and a horizontal line such as ab in Fig. 20 will represent the increase of volume (vaporization) at constant pressure. The point b may be assumed to represent the volume of one pound of dry saturated vapor at a pressure P_1. Obviously the steam, as it is formed, does work in driving out the piston against the resistance offered by the weights which must be raised.

If a stop is provided which will prevent the movement of the piston beyond the position corresponding to the point b, it will be possible to remove the larger weight when that point is reached and the high pressure steam will hold the piston and rod hard against the stop. If now some cooling medium is applied, such as a large piece of ice held against the conducting head of the cylinder or water running over that head, heat will be abstracted and a partial condensation of the steam within the cylinder will occur. As condensation progresses the pressure will drop because there will be less and less steam, by weight, in a given volume. Such a process, would be indicated by the line bc which represents a drop of pressure, while the volume

contained within the cylinder walls between head and piston remains constant.

When some point c is reached, the steam pressure will have been reduced to a value equal to that exerted by the small weight, and the piston will be driven in toward the cylinder head while the heat absorbing medium continues to remove heat from the steam and to cause further condensation. The combination of piston motion and heat absorption will be so regulated that the pressure remains constant at P_2 during this process, because the weight will move the piston inward just as fast as necessary to maintain a constant pressure. If sufficient heat is absorbed, the pound of material within the cylinder will ultimately all be condensed or liquefied and will just fill the volume V_d.

The heat absorbing body may now be removed and an infinitesimal motion of the piston toward the head would serve to raise the pressure on the liquid water from P_2 to P_1 so that the volume V_d may be taken equal to the volume V_a and the line da may be assumed to be vertical. It would then represent an increase of pressure at constant volume. This might be caused by hanging a weight of the larger size on the wire when condition d was reached.

Having brought the material, or working substance, back to the conditions originally shown at a, the high temperature source of heat can again be brought in contact with the end of the cylinder and the entire cycle carried through once more. There is obviously no reason why it could not be repeated as often as desired.

33. Work Done by the Engine. If the device just described is to serve as a steam engine, it must actually make mechanical energy available, that is, it must convert into mechanical form some of the heat energy supplied it. It is now necessary to see whether it does so.

Water vaporizing and increasing in volume as from V_a to V_b was shown in the last chapter to do work upon the piston confining it. Work has been shown to be equal to

(total force×total distance) and in this case if L represents the distance in feet traveled by the piston, the work done by the steam upon the piston while the latter moves from a' to b' must be

$$\text{Work done on piston} = \text{total force} \times \text{distance}$$
$$= P_1 \times \text{area of piston} \times L \ . \ . \ . \ \text{ft.-lbs.}$$

But the product of area of piston in square feet by distance traveled in feet is equal to the piston displacement or volume swept through by the piston, that is $(V_b - V_a)$ cubic feet. Therefore

$$\text{Work done on piston} = P_1(V_b - V_a) \text{ ft.-lbs.} \quad . \quad (19a)$$

$$= \frac{P_1(V_b - V_a)}{778} \text{ B.t.u.} \quad . \quad (19b)$$

The first form of this expression $P_1(V_b - V_a)$ is very obviously represented by the area under the line ab in Fig. 20 and this area therefore represents the work done by the steam upon the piston during the change of volume at constant pressure represented by that line. While the steam is supplying this amount of energy to the piston or doing this amount of work upon the piston, the latter does an equivalent amount of work upon the weights if frictionless mechanism be assumed. In such a case the total weight hung on the wire multiplied by the distance raised would therefore give the same result in foot-pounds as that just obtained.

It should be noted that Eq. (19) is merely an expression of the external work done during vaporization, that is, an expression of the amount of heat which is used for the doing of external work. It is the exact equivalent of the external latent heat previously discussed. In fact, the group of symbols APu is really a condensation of Eq. (19) formed by putting A for $\frac{1}{778}$ and u for $(V_b - V_a)$.

The line *cd* also represents a change of volume at constant pressure and the same type of formula as applied to *ab* will express the work done during this process. In this case, however, the piston is being pushed into the cylinder by the small weight against the pressure of the steam, and energy is being supplied to push the piston in. This energy is equal to the weight of the small weight (pounds) multiplied by the distance it falls (feet). The piston is therefore doing work upon the steam, and the amount is

$$\text{Work done on steam} = P_2(V_c - V_d) \text{ ft.-lbs.} \quad . \quad . \quad (20)$$

$$= \frac{P_2(V_c - V_d)}{778} \text{ B.t.u.} \quad . \quad . \quad (21)$$

The first form of expression also represents the area under the line *cd* and this area therefore represents the work done by the piston upon the steam mixture in the cylinder during the process represented by *cd*.

No work can be done by steam on piston or by piston on steam during the processes represented by *bc* or *da* because both the weights and the piston are stationary during these changes and it has already been shown that work involves motion.

The total work done upon the piston by the steam is therefore represented by the area *abef* and this amount of energy is used in raising the two weights through a vertical distance equal to the piston travel. Some of this energy, or its equivalent, will have to be returned an instant later, however, in order that the piston may do the work shown by the area *cdfe* upon the steam. It is returned by the small weight dropping through a distance equal to the travel of the piston. The net mechanical energy made available by carrying through the series of processes is therefore represented by the area $(abef) - (cdfe) = (abcd)$ or the area enclosed by the four lines representing the

pressure and volume changes experienced by the working substance during one cycle of events. It is equal to the work done in raising the larger weight a vertical distance equal to the travel of the piston.

This net energy made available is obviously

$$\text{Energy made available} = P_1(V_b - V_a) - P_2(V_c - V_d)$$
$$= (P_1 - P_2)(V_b - V_a) \text{ ft.-lbs.} \quad (22)$$
$$= \frac{(P_1 - P_2)(V_b - V_a)}{778} \text{ B.t.u.} \quad (23)$$

Since this amount of energy is made available while one cycle of events is being carried out and since the cycle can be repeated time after time if sufficient heating and cooling mediums are available, any quantity of mechanical energy can be produced from heat energy by repeating the cycle a sufficient number of times. This would correspond to picking up a number of the larger weights which were slid on to the wire at the lower elevation and slid off at the higher.

This repetition of cycles would correspond, in a real engine, to running at such a speed that the required number of cycles would be produced in a given time to make available the amount of mechanical energy required.

Or, the power made available per cycle could be increased. This is easily seen by an inspection of Eq. (22). Increasing the value of either of the right-hand terms will obviously increase the amount of energy made available. The value of $(P_1 - P_2)$ can be increased by raising the initial pressure P_1 or by lowering the final pressure P_2. The value of $(V_b - V_a)$ may be increased by using more than one pound of material, thus increasing both the volume V_b of the saturated steam formed and increasing the volume V_a of the liquid water, but getting a greater numerical value for $(V_b - V_a)$. This would correspond in a real case to using a larger cylinder and therefore a larger engine.

ILLUSTRATIVE PROBLEM

An engine of the type described is to work with a maximum pressure of 100 lbs. per square inch absolute and a minimum pressure of 15 lbs. per square inch absolute. The cylinder is to be of such size that 1 lb. of water is used and the steam is to be dry and saturated at the point b of the cycle.

Find: (a) the amount of mechanical energy made available per cycle; (b) the amount of energy made available per minute if 150 cycles are produced per minute; and (c) the horse power of the engine.

It will first be necessary to find the piston displacement required and the space necessary between piston and cylinder head to accommodate the pound of water in liquid form. The steam tables give the volume of one pound of dry saturated steam at 100 lbs. per square inch as 4.429 cu.ft. and the volume of one pound of water may be taken as 0.017 cu.ft. The values of the various volumes and pressures will therefore be

$$V_a = V_d = 0.017 \text{ cu.ft.};$$
$$V_b = V_c = 4.429 \text{ cu.ft.};$$
$$P_a = P_b = 100\times 144 = 14{,}400 \text{ lbs. per sq.ft.};$$
$$P_c = P_d = 15\times 144 = 2160 \text{ lbs. per sq.ft.}$$

(a) Using Eq. (22) the amount of mechanical energy made available per cycle will be

$$\begin{aligned}(P_1-P_2)(V_b-V_a) &= (14{,}400-2160)(4.429-0.017)\\ &= 12{,}240\times 4.412;\\ &= 54{,}002.88 \text{ ft.lbs.}\end{aligned}$$

(b) If 150 cycles are produced per minute, the total amount of mechanical energy made available per minute must be

$$150\times 54{,}002.88 = 8{,}100{,}300 \text{ ft.-lbs.}$$

(c) The horse power must then be

$$\text{h.p.} = \frac{8{,}100{,}300}{33{,}000} = 245+.$$

34. Heat Quantities Involved. It is a very simple matter to determine the quantity of heat which must be supplied to produce the process ab, and the quantities of

heat which must be removed to produce the processes *bc* and *cd*. This can be done by making use of the known properties of water and steam as given in the steam tables.

The water at *d* must be at the temperature of vaporization corresponding to pressure P_2 since it has just been formed by condensation from steam under that pressure. It therefore contains the heat of the liquid corresponding to that pressure. If it is to be vaporized at pressure P_1, it must first be raised to the higher temperature corresponding to that pressure. The amount of heat required to do this will obviously be the difference between the heat of the liquid at the temperature corresponding to P_1 and the heat of the liquid at the temperature corresponding to P_2. These can be found in the steam tables.

The latent heat of vaporization at P_1 must then be added to cause the increase of volume shown by *ab*. This can also be found in the steam tables for any given case.

The quantity of heat which must be removed to produce the processes represented by *bc* and *cd* can be found similarly from steam table values, although the exact method of procedure is not quite as obvious as in the preceding cases.

Assuming that it is possible to find the heat supplied, Q_1, and the heat removed, Q_2, it is obvious that the energy made available in mechanical form, per cycle, must be equal to (Q_1-Q_2) B.t.u., since this is the amount of heat energy which has disappeared and since it cannot have been destroyed. This may be put in the form of an equation, thus

$$\text{Energy made available} = Q_1 - Q_2. \quad . \quad . \quad (24)$$

If the proper substitutions are made in this formula and it is then simplified, it becomes

$$\text{Energy made available} = (APu)_{P_1} - x_c(APu)_{P_2} \text{ B.t.u.}, \quad (25)$$

in which

$(APu)_{P_1}$ = the external latent heat at pressure P_1;
$(APu)_{P_2}$ = the external latent heat at pressure P_2, and
x_c = quality at point c, which can be found from the ratio of dc to dc'.

Numerical substitution in this equation for any given case will show that it gives exactly the same values as would be obtained by the use of Eq. (23).

It is to be noted particularly that the energy made available is actually less than the *external* latent heat at the higher pressure, while the heat supplied must be equal to the *total* latent heat plus some of the heat of the liquid. An inspection of the steam tables will show that the external latent heat for ordinary steam pressures forms a very small fraction of even the total latent heat, and therefore the mechanical energy made available for a given expenditure of heat energy is very small in the case under discussion.

35. Efficiency. The term efficiency is used in engineering as a measure of the return obtained for a given expenditure. It may be defined in any one of the following ways:

$$\text{Efficiency} = \frac{\text{Useful result}}{\text{Expenditure made to obtain that result}}$$

$$= \frac{\text{Result}}{\text{Effort}}$$

$$= \frac{\text{Output}}{\text{Input}} \quad \ldots \ldots \ldots \ldots \quad (26)$$

In the case of a heat engine, the useful result is the mechanical energy obtained by the operation of the engine, while the expenditure made is the heat which is supplied. For this case efficiency may therefore be defined by the expression

$$\text{Engine efficiency} = \frac{\text{Mechanical energy obtained per cycle}}{\text{Heat supplied per cycle}}$$

$$= \frac{E}{Q_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

$$= \frac{Q_1 - Q_2}{Q_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

in which

E stands for mechanical energy obtained,
Q_1 stands for heat supplied, and
Q_2 stands for heat rejected.

In the case of the type of steam engine just considered, this efficiency would have a value between 6 and 8 per cent for ordinary pressures. That is, the engine would produce in mechanical form only 6 to 8 per cent of the energy supplied it in the form of high temperature heat. Moreover, these figures would hold only for a theoretically perfect engine; a real engine built to operate upon this cycle would probably give efficiencies of the order of 2 to 3 per cent. The reasons for this great discrepancy will be discussed in a later chapter.

36. Effect of Wet Steam. In what has preceded, it was assumed that the pound of steam was completely vaporized along the line *ab* so that dry, saturated steam existed in the cylinder at *b*. It might, however, be assumed that vaporization was incomplete at the upper right-hand corner of the cycle, so that this corner occurred at a point to the left of *b* in Fig. 20 and with a quality x less than unity.

Under such conditions, the cylinder would not have to be so big, since the maximum volume attained by the steam would be smaller than in the preceding case. The work done per cycle would obviously be smaller in quantity, because the area enclosed within the lines of the cycle would be smaller. It can also be shown that the efficiency would be lowered by incomplete vaporization.

37. Application to a Real Engine. The engine which has been described in the preceding paragraphs could easily be converted into the counterpart of a real engine by substituting connecting rod, crank shaft and flywheel for wire, pulley and weights as described in the first paragraph of this chapter. It could then be made to do work in just the same way as has been described; some of the energy made available during the outstroke would be used for overcoming resistance at the shaft, that is, doing useful work, and some of it would be stored in the flywheel which would speed up slightly. The energy which must be expended on the steam during the return stroke would be obtained by allowing the flywheel to slow down and thus deliver sufficient kinetic energy to drive the piston back against the low-pressure steam. The cycle and the efficiency would thus, theoretically, be exactly the same as those just investigated.

Great difficulty would, however, be met in a real engine if the steam had to be formed and condensed within the cylinder, and another method which gives the same results is therefore used. Steam is generated in a boiler and allowed to flow into the cylinder and push out the piston just as though it were actually being formed in the cylinder as previously described. When the piston reaches the end of its outstroke the inlet valve is closed and the exhaust valve is opened, allowing some of the steam to blow out into a space in which a lower pressure exists. As the piston stands still at the end of its stroke while the pressure drops, the line *bc* is produced as in the previous description, but by a different method. The piston then returns and drives the remaining steam out of the cylinder at a constant pressure theoretically equal to that of the space into which the steam is being forced or exhausted. The line *cd* is thus produced and the closure of the exhaust valve and opening of the admission valve when *d* is reached will start the cycle over again.

In order to get more work out of a given size of cylinder and to obviate the necessity of giving back energy which has already been given out, engines are generally made to take steam on both sides of the piston. They are then known as **double acting engines.** In this case the steam admitted on one side of the piston would supply the energy necessary both for overcoming the resistance due to the load and for driving out the low-pressure steam on the other side of the piston. On the return stroke conditions would be just reversed.

38. Desirability of Other Cycles. The cycle of operations described in preceding paragraphs is the most inefficient of all those actually used, that is, it gives the smallest return for a given amount of heat supplied. This is because only the external latent heat supplied is converted into mechanical energy and part of that energy must be returned to complete the cycle. All of the heat of the liquid as well as all the internal latent heat supplied along *ab* passes through the engine without conversion and is exhausted.

Therefore, cycles which differ from that described in such a way as to make it possible to convert into mechanical energy some of the internal latent heat and possibly some of the heat of the liquid should be highly desirable as they ought to yield a larger return of mechanical energy for the same total amount of heat supplied. Two such cycles are commonly used; they may be described as the *Complete-expansion cycle* and the *Incomplete-expansion cycle.* The former is used in steam turbines, the latter in most reciprocating steam engines. The rectangular cycle which has just been described is used in duplex pumps and similar apparatus.

39. The Complete-expansion Cycle. This cycle, which is also known as the Clausius and as the Rankine cycle, starts just the same as that already described. This is shown in Fig. 21. The pressure on, say, a pound of water

is raised from P_2 to P_1 and its temperature is raised from that of vaporization at P_2 to that of vaporization at P_1. After this it is vaporized, giving the increase of volume

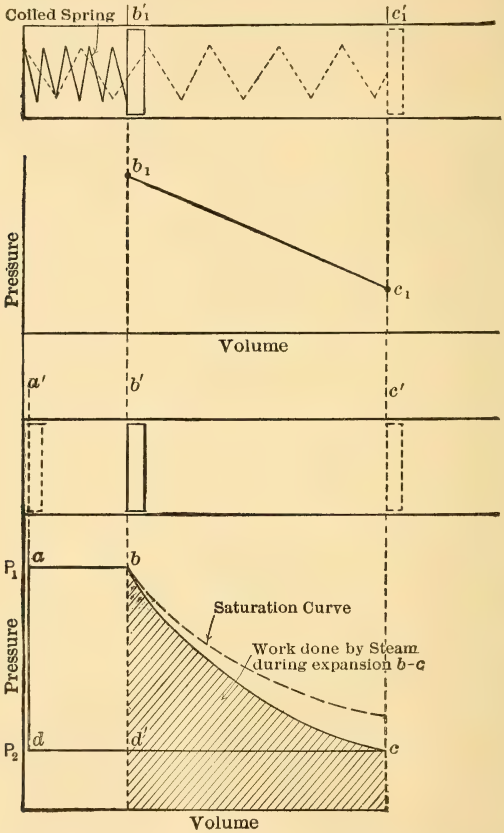


FIG. 21.—Complete Expansion Cycle or Clausius Cycle.

shown by ab. The supply of heat is then stopped. The cylinder of the engine is made larger than in the preceding type so that when the point b' is reached the piston can travel still further, and it is allowed to do so, that is, the

high-pressure steam is allowed to push it further out. This can be pictured by imagining the steam to act like the compressed spring shown in the figure and to push the piston in much the same way as does the spring. The line b_1c_1 shows the decreasing pressure exerted on the piston by the spring as the latter expands so as to get longer and longer. Because of the properties of a spring this is a straight line. The line bc shows the decreasing pressure exerted on the piston by the steam as the latter expands so as to occupy greater and greater volumes. Because of the properties of steam this line is curved instead of straight.

Work will be done on the piston by the expanding steam during the process bc and the amount of this work will be indicated by the area under the line bc as shown in the figure. This work must have been done by the expenditure of energy on the part of the steam and since no energy was added after the point b was reached the work must have been done at the expense of heat energy contained in the steam at b. It has already been shown that the heat above 32° in the steam at b is equal to the sum of the heat of the liquid and the internal latent heat, and some of this heat must obviously be used for the doing of work along bc instead of being entirely rejected to the cooling medium as in the preceding cycle without "expansion."

The expansion of the steam continues until the "back pressure" P_2 is reached. The cooling medium may then be imagined to be brought into use and to abstract such heat of vaporization as may remain in the steam besides absorbing the equivalent of the work done on the steam by the returning piston, thus giving the process shown by the line cd.

If the expansion line bc of the cycle just described could be carried out within walls constructed of such material that it would not give heat to nor take heat from the steam, it is obvious that any heat energy lost by the steam during the expansion could be lost only by conver-

sion into mechanical energy. An expansion of this kind is called an **adiabatic expansion.**

In the figure, the curve of adiabatic expansion is shown in its correct position with respect to the saturation curve and it is obvious that *for an adiabatic expansion, starting with dry, saturated steam, the quality decreases as the expansion progresses.*

Comparison with Cycle without Expansion. The heat supplied is the same in both of the cycles just considered when they operate between the same two pressures, but the mechanical energy obtained in the case of the complete expansion cycle is much greater. In Fig. 21, for instance, the mechanical energy obtainable with the cycle first described is represented by the area *abd′d* while that obtainable with the complete expansion cycle with the same heat supply Q_1 is represented by the same area *abd′d* plus the additional area *bcd′*. The efficiency of the complete expansion cycle is therefore very much higher than that of the cycle without expansion.

For conditions similar to those giving a theoretical efficiency of about 6 per cent without expansion, the complete expansion cycle will give a theoretical efficiency of about 12 per cent and this figure can be doubled by expedients which will be considered later.

The cylinder required for the production of the complete expansion cycle would be much larger than that required for the other cycle if both used the same weight of steam per cycle. The proportion would be in the ratio of the volume shown at *c* in Fig. 21 to the volume shown at *b*. But the complete expansion cycle would make available much more energy per pound of steam than would the other, so that the difference in the size of cylinders would not be so great if both were required to make available the same amount of mechanical energy per cycle.

40. The Incomplete-expansion Cycle. The shape of this cycle is shown in Fig. 22. It is just like the complete

expansion cycle down to the point *c*. The cylinder in which it is produced has a smaller volume than that used for the complete expansion cycle so that the piston arrives at the end of its stroke before it has opened up volume enough to enable the steam to expand all the way down to the lowest pressure (terminal or back pressure). When the point *c* is reached in the real engine, the exhaust valve is opened and enough steam then blows out to reduce the pressure to the back pressure P_d. The piston then returns and drives out the remainder of the steam as shown by the line *de*.

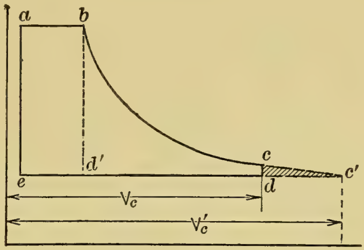


FIG. 22.—Incomplete Expansion Cycle.

In the ideal method assumed in the preceding treatment, the heat absorbing medium would be brought into use at *c*, absorbing sufficient heat to reduce the pressure from P_c to P_d while the piston remained stationary at the end of its stroke. The latent heat of vaporization remaining in the steam at *d* would then be absorbed as the piston was driven back from *d* to *e*.

Comparison with Other Cycles. The incomplete expansion cycle is intermediate between the two previously discussed. This can be appreciated readily by an inspection of Fig. 22. In this figure the area *abd'e* represents the mechanical energy obtainable with the cycle without expansion; the area *abc'e* represents the energy obtainable from the same quantity of steam with complete expansion; and the area *abcde* represents the energy obtainable from the same amount of steam with incomplete expansion.

The later the point at which the exhaust valve is opened, point *c*, the more nearly do efficiency and energy obtainable approach the values for the complete expansion cycle.

The earlier the point at which the exhaust valve is opened, the more nearly do efficiency and energy obtainable approach the values for no expansion.

Despite the lower efficiency of the incomplete expansion cycle as brought out in connection with Fig. 22 it is universally used on all reciprocating engines excepting those which make no pretense to economy and use no expansion. The less efficient cycle is used for the simple reason that complete expansion in a reciprocating engine does not pay commercially. For complete expansion the cylinder must be larger in the ratio of V_c to $V_{c'}$ as shown in Fig. 22 and the work obtained by completing the expansion is a very small part of the total. In most cases it would not be great enough to overcome the friction of the engine, not to mention paying interest on the necessarily higher cost of the larger cylinder and accompanying parts.

It will be shown in a later chapter that the steam turbine can economically expand the steam completely and the complete expansion cycle is therefore used with such prime movers.

CHAPTER V

ENTROPY DIAGRAM

41. Definitions. In Chapter III temperature, pressure and volume were discussed as criteria determining the condition of water and steam. Other things may be used in determining the condition of such materials. One which is particularly useful from an engineering standpoint is known as *entropy* and is designated by the Greek letter ϕ.

For every condition of water and steam, there is a characteristic value of entropy just as there is a characteristic value of temperature, pressure, volume, heat above 32° F., etc. These values of entropy are given in the steam tables in just the same way as the value of temperature, pressure, volume, heat above 32° F., and such, are given.

The entropy of the liquid given for any particular pressure is the change of entropy experienced by one pound of the liquid when its temperature is raised from 32° F. to the temperature of vaporization corresponding to that particular pressure. It might be spoken of as the entropy of the liquid above 32° F., just as q is spoken of as the heat of the liquid above 32° F. It is represented by ϕ_l.

The entropy of vaporization given for any particular pressure is the change of entropy experienced by one pound of the material while changing from water at the temperature of vaporization to dry saturated steam at constant pressure. It corresponds to the latent heat of vaporization and is designated by ϕ_v.

The entropy of dry saturated steam at any pressure is the sum of ϕ_l and ϕ_v and therefore is the total change of entropy experienced by a pound of material in changing

from water at 32° F. to dry saturated steam at the particular pressure in question.

The entropy of superheat at any pressure and temperature is the change of entropy experienced by a pound of dry, saturated steam at that pressure when superheated to that particular temperature. It is designated by ϕ_s.

The entropy of superheated steam at any pressure and temperature is the total change of entropy experienced by one pound of material when changed from water at 32° F.

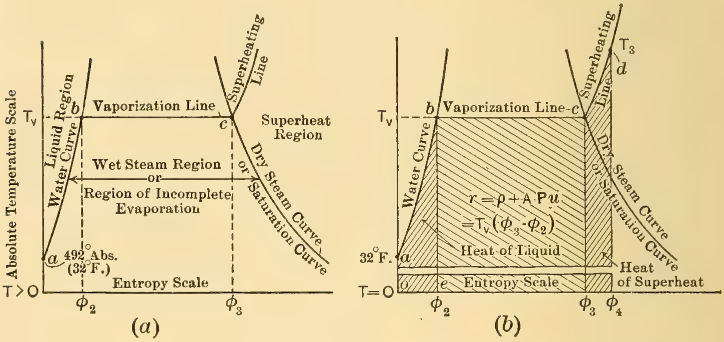


FIG. 23.—Temperature-Entropy Diagrams.

to superheated steam at the pressure and temperature in question. It is equal to $\phi_l + \phi_v + \phi_s$.

42. Temperature-Entropy Chart for Steam. Entropy is particularly useful to the engineer because it enables him to draw charts which lend themselves readily to an easy, graphical solution of certain problems which would otherwise involve complex calculations. One of these charts is known as the *Temperature-Entropy Chart.*

In making this chart, absolute temperature is generally plotted on the vertical and entropy above some datum temperature on the horizontal, as shown in Fig. 23 (*a*) and (*b*), which represents the construction of a temperature entropy diagram for water and steam. The entropy values on

this chart are plotted above 32° F. as datum temperature.

The water line or water curve is obtained by picking out of the steam tables the values of ϕ_l, entropy of the liquid, for different pressures and plotting them against the absolute temperatures corresponding to those pressures. Obviously, zero of entropy will occur at the absolute temperature corresponding to 32° F., i.e., about 492° F. abs.

The saturation curve or **dry steam curve** is obtained by picking out of the steam tables the values of $\phi_l+\phi_v$ for different pressures and plotting against corresponding absolute temperatures.

The entropy of vaporization is obviously shown for each different temperature (or pressure) by the distance between the water curve and the saturation curve, since the former is distant from the vertical axis by an amount equal to ϕ_l, while the latter is distant an amount equal to $\phi_l+\phi_v$.

Superheating lines are drawn by picking from the steam tables the values of entropy above 32° F. for steam superheated to different temperatures at one particular pressure and plotting against the proper temperatures. There will be as many superheating lines on the diagram as one chooses pressures for which to plot them. Only one is shown in the figure.

One very useful property of this diagram follows from the fact that points on its surface indicate the condition of the material. For instance, if the temperature-entropy, or $T-\phi$, values of the material at a given condition should plot to the left of the liquid line, the material must be in the liquid condition; if they plot between the liquid line and the saturation curve, the material must be a mixture of liquid and saturated vapor; if they plot on the saturation curve, the material must be dry, saturated steam; and if they plot to the right of the saturation curve, the material must be superheated steam. This all follows directly from the definition of entropy above 32° F., as plotted in these dia-

grams. The various regions, or fields, into which the diagram divides in this way are shown in Fig. 23 (*a*).

Another very useful property of this diagram follows from the fact that area represents heat just as area on a pressure-volume diagram was found to represent work. Thus the area under the line *ab*, for instance, represents the heat required to raise the temperature of one pound of water from 32° F. to the temperature at *b*. Similarly the area under the line *bc* represents the heat required to change a pound of water at the temperature at *b* to a pound of dry, saturated steam at the same temperature. The heat required to superheat this pound of saturated steam at constant pressure up to the temperature shown at *d* is similarly represented by the area under the line *cd*.

In this connection, it should be noted that this diagram is plotted above absolute zero of temperature just as the pressure-volume diagram is plotted above absolute zero of pressure. The areas in question therefore extend down to the absolute zero of temperature. In order to indicate this in Fig. 23 (*b*), a large part of the chart is supposed to have been broken out, so that the lower end of the diagram could be moved up into view. In Fig. 23 (*a*), the bottom of the diagram is drawn a few degrees below 32° F. and this is indicated by putting $T > O$ opposite the horizontal axis.

The various areas hatched in Fig. 23 (*b*) indicate the various quantities of heat previously discussed. It should be understood that the areas represent the heat quantities only for the particular pressure which corresponds to the temperature indicated by T_v. For a higher pressure, the line *bc* would be higher and the areas proportionately larger; for a lower pressure the line *bc* would be lower and the areas smaller.

ILLUSTRATIVE PROBLEM

Starting with liquid at a temperature T_1 corresponding to the temperature of vaporization at a pressure of 50 lbs. per square inch absolute, assume the liquid raised to the temperature of vaporization at a pressure of 100 lbs. per square inch absolute and then completely vaporized. Determine the various changes of entropy and indicate them on a $T\phi$-chart.

The steam tables give entropy of the liquid, ϕ_l, as equal to 0.4113 for water about to vaporize under 50 lbs. per sq. in. absolute, and 0.4743 for water about to vaporize under a pressure of 100 lbs. per sq. in. absolute. The difference, that is, $0.4743-0.4113=0.0630$, must be the entropy change experienced by the liquid when its temperature is raised from the lower to the higher value. These values are shown in Fig. 24.

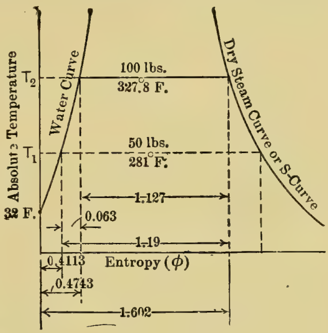


FIG. 24.

The steam tables give entropy of vaporization, ϕ_v, at 100 lbs. per square inch absolute as 1.1277. Adding this to the entropy above 32° F. of the liquid at vaporization temperature under 100 lbs. pressure gives $0.4743+1.1277=1.602$ as the entropy above 32° of dry, saturated steam at 100 lbs. per square inch absolute. These values are all indicated in their proper position in Fig. 24.

The total change of entropy experienced by the material in changing from water at the temperature of vaporization under 50 lbs. pressure to dry, saturated steam at 100 lbs. pressure is obviously equal to $0.0630+1.1277=1.1907$.

43. Quality from Tϕ-chart. The entropy change experienced by steam in the process of vaporization is directly proportional to the addition of heat. Thus, when half the latent heat has been added to one pound of material, the entropy change is $\frac{1}{2}\phi_v$. In general, if a fraction x of the latent heat has been added, the entropy change has been $x\phi_v$ during the process. Therefore, if the temperature entropy condition of a pound of material should plot at a

point such as c in Fig. 25, it follows that the material is a mixture of water and steam and that a fraction of the pound equal to $\frac{bc}{bd}$ is steam, the rest being water. But, by definition, the fraction $\frac{bc}{bd}$ is x, the quality of the material.

The temperature-entropy chart is very useful when used in connection with this property of showing quality. Thus, in Fig. 25, the area under bc, down to absolute zero temperature, represents the fraction of the latent heat of

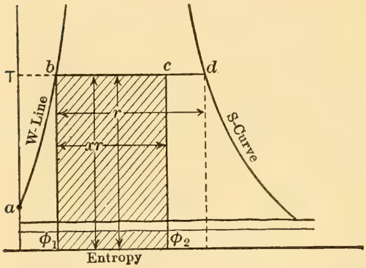


FIG. 25.—Quality from Temperature-Entropy Chart.

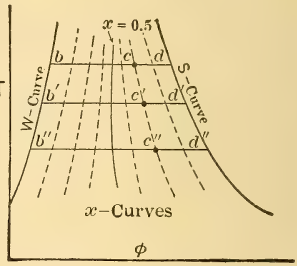


FIG. 26.—Constant Quality Curves.

vaporization per pound which must be added to give a pound the quality x.

For convenience in use, **constant quality lines** are generally drawn on temperature-entropy charts. Such lines are shown in Fig. 26. Each line is obtained by plotting the temperature entropy conditions for a given quality at different pressures. For this purpose, ϕ_v and ϕ_l are taken from the steam tables for a given pressure. The numerical value of ϕ_v is then multiplied by the fraction representing the chosen quality, say 0.9, and the product is added to ϕ_l, giving the total entropy above 32° F. for quality 0.9 at the particular pressure chosen. The same

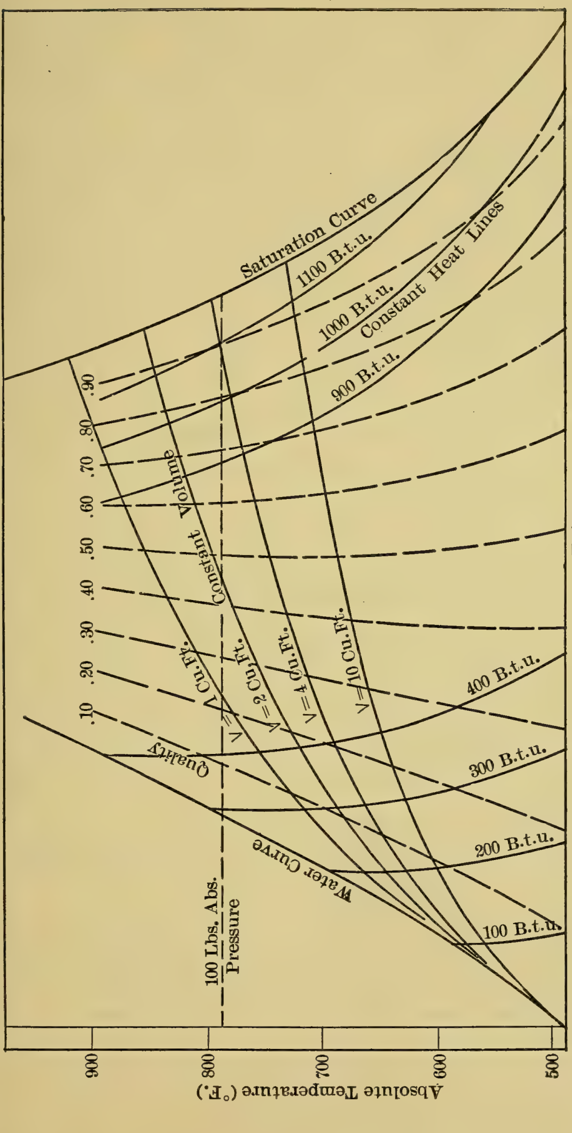


Fig. 27.—Constant Volume, Constant Quality, and Constant Heat Lines.

process is repeated with the same value of the quality, but with different pressures, until enough points have been secured to make it possible to draw a smooth line through them.

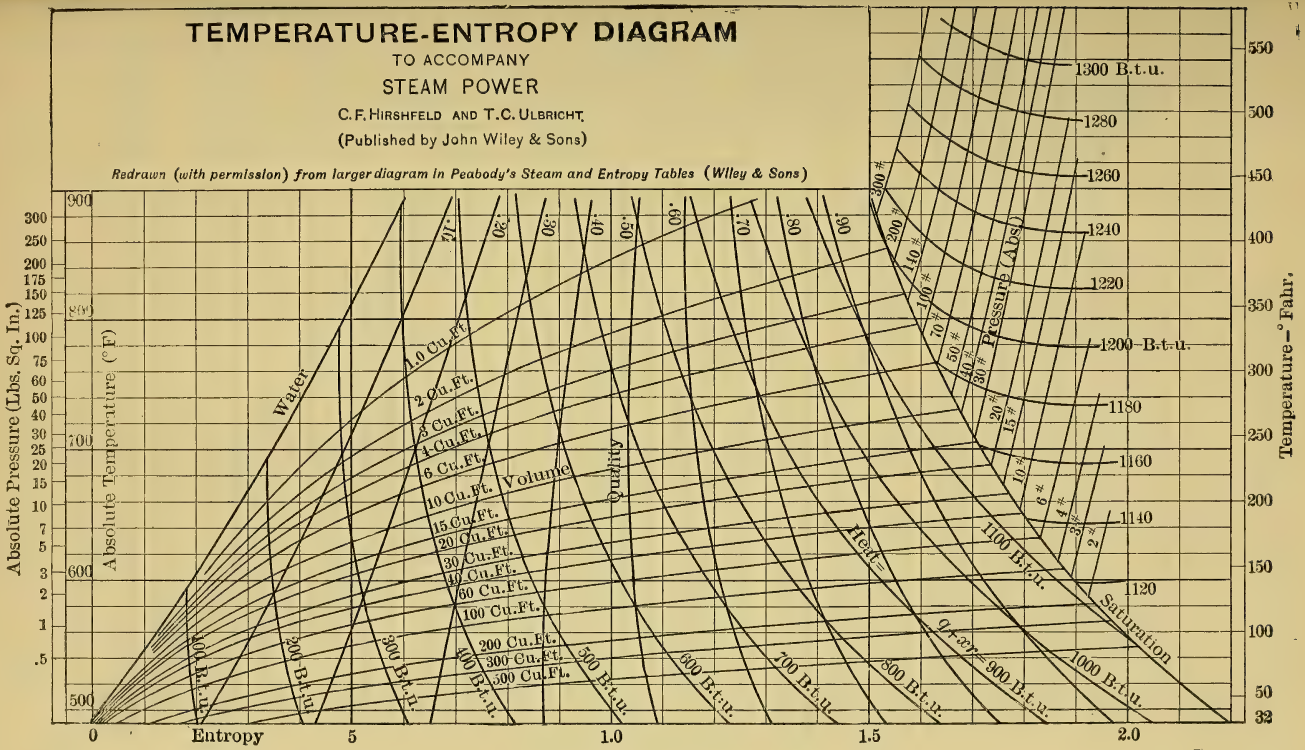
44. Volume from Tϕ-chart. Since quality changes at any given temperature, or pressure, are accompanied by volume changes, it is possible to find a series of values for the quality of a pound of wet steam which will make that pound occupy the same volume at different temperatures. Having found the quality which will be necessary at a number of different temperatures, the total entropy above 32° F. can be found for each case and these values can then be plotted on the $T\phi$-chart. Connecting the points so obtained would give what is known as a **Constant Volume Line.**

Several of these constant volume lines are shown in their correct positions in Fig. 27. It will be observed that, for each volume, the quality must increase as temperature (and pressure) increases in order to maintain a constant value for the volume occupied by one pound of mixture.

45. Heat from Tϕ-chart. Equations for obtaining the total heat above 32° F. for wet and for superheated steam were given in an earlier chapter. By means of these equations, it is possible to find a succession of values for quality and superheat which will give a pound of material any chosen heat content at different pressures. If the corresponding values of temperature and entropy are found and plotted, what is known as a **Constant Heat Line** results. Several of these lines are shown in Fig. 27.

46. The Complete Tϕ-chart for Steam. A very complete, graphical representation of the properties of water and steam can be procured by combining in one diagram all of the lines discussed in preceding paragraphs. Such a diagram is generally spoken of as the *$T\phi$-diagram* or the *$T\phi$-chart* for steam. An example of such a diagram is given in Fig. 28.

This chart is very useful, as it enables one to solve by



TEMPERATURE-ENTROPY DIAGRAM
TO ACCOMPANY
STEAM POWER
C. F. HIRSHFELD AND T. C. ULBRICHT.
(Published by John Wiley & Sons)
Redrawn (with permission) from larger diagram in Peabody's Steam and Entropy Tables (Wiley & Sons)
Absolute Pressure (Lbs. Sq. In.)
Absolute Temperature (° F)
Temperature—° Fahr.
Entropy
Water
Quality
Volume
Pressure (Abs.)
Heat = q + xr
Saturation
1300 B.t.u.
1280
1260
1240
1220
1200 B.t.u.
1180
1160
1140
1120
1100 B.t.u.
1000 B.t.u.
900 B.t.u.
800 B.t.u.
700 B.t.u.
600 B.t.u.
500 B.t.u.
400 B.t.u.
300 B.t.u.
200 B.t.u.
100 B.t.u.
1.0 Cu.Ft.
2 Cu.Ft.
3 Cu.Ft.
4 Cu.Ft.
6 Cu.Ft.
10 Cu.Ft.
15 Cu.Ft.
20 Cu.Ft.
30 Cu.Ft.
40 Cu.Ft.
60 Cu.Ft.
100 Cu.Ft.
200 Cu.Ft.
300 Cu.Ft.
500 Cu.Ft.
.10
.20
.30
.40
.50
.60
.70
.80
.90
300 #
200 #
140 #
100 #
70 #
50 #
40 #
30 #
20 #
15 #
10 #
6 #
4 #
3 #
2 #

FIG. 28.

inspection many of the most difficult problems which arise in the theory and practice of using steam. As an example, assume that it is desirable to know what will happen if water at the temperature of vaporization corresponding to about 24 lbs. per square inch absolute has its volume increased indefinitely at constant temperature. The initial condition of the water would be shown on the water curve of Fig. 28 at the point at which the 700° absolute temperature line crosses it. Increase of volume at constant temperature would be indicated by a horizontal line running to the right from this point. Obviously, vaporization will occur at constant pressure (because the temperature is constant) and the quality will change from zero to unity at which the saturation curve will have been reached. Further increase of volume can result only in the production of superheated steam, since the line representing the process will run out into the superheated steam field. It is also interesting to note that the pressure on the material will have to be decreased as the volume increases in the superheated steam region, as is evidenced by the fact that the horizontal line representing the assumed process cuts lower and lower pressure lines as it is extended to the right in the superheated field.

Note also that the intersections of this horizontal line with constant volume and constant heat lines afford the means of determining volume and heat above 32° F. at different stages of the assumed process.

PROBLEMS

1. Determine from the steam tables the change of entropy experienced by one pound of water when its temperature is raised from 32° F. to the temperature of vaporization under a pressure of 100 lbs. per square inch absolute.

2. Determine from the steam tables the entropy change experienced by one pound of water when its temperature is raised from 32° F. to the temperature of vaporization under a pressure of 150 lbs. per square inch absolute.

3. Determine the entropy change experienced by one pound of water when its temperature is raised from the temperature of vaporization corresponding to 100 lbs. per square inch to that corresponding to 150 lbs. per square inch by subtracting the value found in Prob. 1 from that found in Prob. 2.

4. Determine the change of entropy experienced by one pound of material completely vaporizing at a temperature of 327.8° F.

5. Plot a $T\phi$-chart for one pound of water. Start by plotting entropy of the liquid for various temperatures; then plot entropy of saturated steam (above 32° F.); finally draw water line, saturation line, and several lines showing change of entropy during vaporization.

6. Determine from a $T\phi$-chart the quality which would be attained by one pound of steam if it experienced a change which carried it from the condition of dry saturated steam at 150 lbs. per square inch absolute to a pressure of 25 lbs. per square inch absolute by a process which would plot as a vertical line on the $T\phi$-chart.

7. Assume a pound of mixed water and steam to have a quality of 80% at a pressure of 200 lbs. per square inch absolute. Determine from the $T\phi$-chart the heat above 32° per pound of mixture and the volume occupied by the mixture. Determine also the quality attained if the pressure of the material drops to 20 lbs. per square inch absolute at constant entropy. How does the heat above 32° F. change during such a process?

8. Assume a pound of mixture as in Prob. 7, but with a quality of 30% at a pressure of 200 lbs. Find all quantities called for in that problem.

9. Assume a pound of material as in Probs. 7 and 8 above, but superheated 200° at a pressure of 200 lbs. per square inch absolute. Determine all quantities called for in Prob. 7.

10. Choose a point on the $T\phi$-chart at which a constant volume line intersects the saturation curve. Determine the change of quality, entropy and heat above 32° F., if the material drops to half pressure at constant volume.

CHAPTER VI

TEMPERATURE ENTROPY DIAGRAMS OF STEAM CYCLES

47. Complete Expansion Cycle. This cycle was considered in Chapter IV and the PV-diagram was given there as Fig. 21. The diagram of this cycle drawn to $T\phi$-coordinates is shown in Fig. 29. The same letters are used to represent corresponding points in the two diagrams.

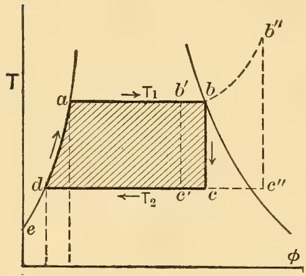


FIG. 29.—$T\phi$-diagram, Complete Expansion Cycle.

The entropy change during the heating of the liquid is shown by the part of the liquid line between d and a, and the heat supplied during that process is represented by the area below the line da, measuring clear down to the absolute zero of temperature.

The entropy change during vaporization is represented by the line ab and the heat supplied during the process is shown by the total area under that line.

The adiabatic expansion of the steam is represented by the line bc, such an adiabatic change fortunately being a constant entropy process and therefore easily drawn in this diagram. Obviously no heat is received or removed during this process, as there is no area under the line bc.

The entropy change during condensation is represented by the line cd and the heat rejected by the working substance during this process is represented by the area under that line.

48. Area of Cycle Representative of Work. It will be remembered that area under a line in the PV-diagram represents work in foot-pounds. That diagram, however, gives no indication of heat received or rejected and it is not possible to obtain any direct idea of efficiency from it. In this respect, the $T\phi$-diagram is much better. Area under the lines da and ab in Fig. 29 represents heat supplied the working substance. Area under the line cd represents heat rejected by the working substance. The difference between these two, or the area enclosed within the lines of the cycle, must therefore represent the heat converted into mechanical energy per cycle.

This diagram therefore shows directly by areas the heat supplied, the heat rejected, and the heat converted into mechanical energy. Further, the ratio of the area representing heat converted into work, and the area representing heat supplied must be the efficiency of the cycle.

Remembering also that if the lines of the cycle are drawn upon a $T\phi$-chart such as that given in Fig. 28, all volume changes, heat contents and qualities at different points are shown without further work, it becomes evident that this form of representation is decidedly convenient and far superior to the pressure volume method.

49. Modifications for Wet and Superheated Steam. The complete expansion cycle is supposed to represent an idealization of what happens in a real prime mover. In real cases, however, the steam may arrive at the prime mover wet or superheated and it is desirable to investigate the method of representing such conditions as well as their effects.

Wet steam corresponds to incomplete vaporization, i.e., a quality less than unity at the upper right-hand corner of the cycle. This might be shown for a given case by the location of the point b' in Fig. 29. The cycle would then be $ab'c'd$ and a smaller amount of work would be obtained per pound of working substance as evidenced by the smaller area enclosed within the lines of the cycle.

In the case of superheated steam, superheating occurs at constant pressure after vaporization is complete. This would be shown by the location of the upper right-hand corner of the cycle at some point b'' on the constant pressure line which extends out from b. The cycle is now represented by $abb''c''d$ and evidently has a different shape than it had in the preceding cases. Obviously the area enclosed within the lines of the cycle is greater than it was before and therefore more mechanical energy is obtained per pound of steam.

50. Incomplete Expansion Cycle. The only difference between the incomplete and complete expansion cycles is

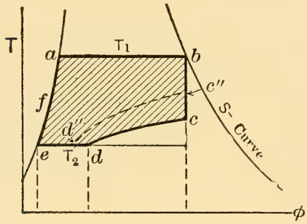


FIG. 30.—$T\phi$-diagram, Incomplete Expansion Cycle.

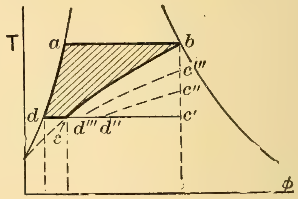


FIG. 31.—$T\phi$-diagram, Cycle Without Adiabatic Expansion.

the termination of the expansion in the former by means of a constant volume line. This is shown to $T\phi$-coordinates in Fig. 30 in which the incomplete expansion cycle is drawn in heavy lines over the one in which expansion continues to the back pressure.

The constant volume line is seen to cut off a corner, thus reducing the area representing heat converted into work. The heat supplied in each case is measured by the area under the lines ea and ab. The efficiency of the cycle with incomplete expansion can therefore be seen to be less than that of the other cycle by simple inspection of the diagram.

If the adiabatic expansion is terminated at a higher

pressure, as by the constant volume line $c''d''$ in Fig. 30, still more of the work area is lost, but the same quantity of heat is supplied, and therefore the efficiency is still lower than when the expansion terminated at c. Obviously as the point at which the adiabatic expansion is terminated moves nearer and nearer to b as shown in Fig. 31, the cycle becomes less and less efficient. If the constant volume line starts at b, there is no adiabatic expansion and the cycle becomes that previously considered as having a rectangular shape in the PV-diagram. This cycle has the shape indicated by $abcd$ in the $T\phi$-diagram of Fig. 31. Obviously it is least efficient of all as was previously shown by other means.

51. Effect of Temperature Range on Efficiency. It has already been stated (see p. 26) that heat engines receive heat at a high temperature, convert some of it into mechanical form and discharge the remainder at a lower temperature. Inspection of the $T\phi$-diagram shows this very clearly, and, remembering that the area of the cycle measures the heat converted, these diagrams also show how raising the upper temperature (or pressure) or lowering the lower temperature (or pressure) will increase the efficiency. It can be seen readily that lowering the lower temperature will, however, be more effective in increasing the efficiency than raising the upper temperature.

PROBLEMS

1. Draw a complete expansion cycle to $T\phi$-coordinates for the following conditions (using $T\phi$-diagram for steam to get values); weight of working substance, 1 lb.; initial pressure, 125 lbs. absolute; quality at beginning of adiabatic expansion, 100% back pressure, 10 lbs. absolute.

2. Determine the following values for cycle drawn in Prob. 1:

(*a*) Entropy of liquid at beginning of vaporization;
(*b*) Entropy at beginning of adiabatic expansion;
(*c*) Quality at end of adiabatic expansion;
(*d*) Volume at end of adiabatic expansion·
(*e*) Entropy at end of condensation.

3. Show by measuring the area on $T\phi$-diagrams, the increase of efficiency resulting from the use of an initial pressure of 175 lbs. absolute and from the use of a terminal pressure of 2 lbs. absolute in place of the values given in Prob. 1.

4. Compare the efficiency of a complete expansion cycle with conditions as in Prob. 1 with a complete expansion cycle with same pressures but with a temperature of 500° F. at the beginning of the adiabatic expansion.

5. Draw an incomplete expansion cycle to $T\phi$-coordinates for the same pressures as in Prob. 1, but with adiabatic expansion ending at a pressure of 15 lbs. absolute.

6. Compare work and efficiency of the two cycles of Probs. 1 and 5 above.

7. Draw a cycle without expansion for the conditions of Prob. 1 to $T\phi$-coordinates and compare the work area with that obtained in Probs. 1 and 5.

CHAPTER VII

THE REAL STEAM ENGINE

52. Operation of Real Engine. In previous chapters the ideal steam engine was considered and several cycles upon which it might be operated were discussed. Real engines are built to operate on the same cycles, but because of certain practical considerations they only imperfectly approximate the ideal performance.

Real engines must be built of iron and steel for practical reasons and these metals absorb, conduct and radiate heat so that certain heat interchanges between the working substance and engine and certain heat losses occur in practical operation. These were eliminated in the ideal case by simply assuming ideal materials not possessed of the characteristics of real metals.

It is also practically impossible to generate steam in the cylinder of a real engine as was assumed to be done in the ideal case. Heat is practically obtained by the combustion of fuels, and the higher the temperature attained the better can the liberated heat be utilized in the generation of steam. To subject the cylinder to such high temperatures and to control the heating and cooling as necessary to produce a number of cycles in rapid succession would lead to rapid wear and great practical difficulties. It has been found best to generate the steam in a boiler which is properly equipped for that purpose and then to transmit it with its contained heat to the engine, which is constructed in such a way as to utilize that heat to the best advantage. If the steam is to be condensed, as assumed in the ideal cases, it has also been found best to remove it from the

cylinder and to condense it in a separate piece of apparatus properly constructed for that purpose.

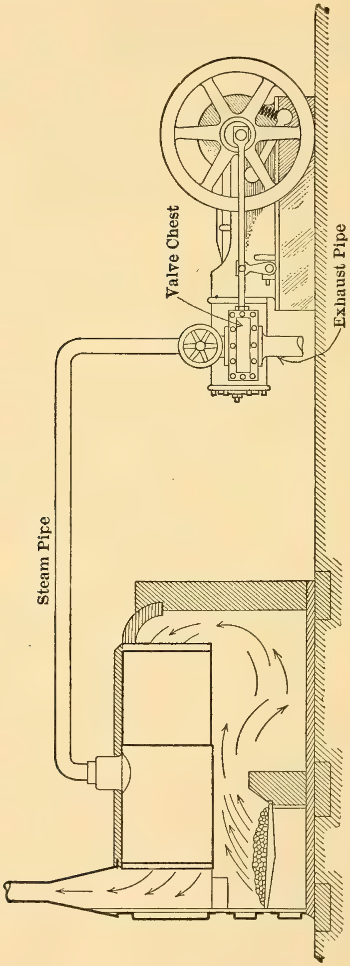


Fig. 32.—Elementary, Non-condensing Plant.

The entire arrangement which results from these practical modifications in the case of a non-condensing engine

is shown in Fig. 32. Steam is generated within the boiler at some constant pressure P_1 and at the proper instant the admission valve at one end of the cylinder is opened, allowing steam to flow in and drive the piston outward. If there were no losses, this would be represented by some such line as ab at a height P_1 on the PV-diagram of Fig. 33. Closing of the valve after the piston had moved part way out would cut off the further flow of steam, and, with continued motion of the piston, the steam within the cylinder would expand. If no heat interchanges occurred, this expansion bc would be adiabatic as in the ideal case.

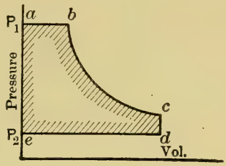


Fig. 33.

It will be observed that the two lines on the PV-diagram thus far produced represent equally well the corresponding two lines of the complete or incomplete-expansion cycles. The heat supplied in the boiler is the same as that supplied in the cylinder under the ideal conditions originally assumed, and the work under the line ab is equal to the external work done during vaporization just as in the ideal case. If difficulty is experienced in connection with the statement regarding external work, it is only necessary to picture the process in this way: Assume that each pound of steam formed in the boiler does the external work equivalent to APu by pushing the pound previously generated ahead of it as a piston, and that this motion communicated along the pipe from layer to layer results in pushing an equivalent weight (and volume) into the cylinder against the resistance offered to the piston's motion.

When the piston arrives at the end of its stroke at the point c, the opening of the "exhaust valve," connecting the interior of the cylinder with the space in which the pressure P_2 lower than P_c is maintained, will permit some of the steam to blow out of the cylinder with the piston standing stationary at the end of its stroke. This would

give a constant volume change similar to the corresponding line in the incomplete-expansion cycle.

The return of the piston from d to e, with the exhaust valve still open, would force the remainder of the steam out of the cylinder and into the space in which the pressure P_2 is maintained. The result, so far as the diagram is concerned, is obviously the same as in the ideal case, and if the steam were condensed within a proper vessel into which it exhausted (instead of being exhausted to atmosphere), the result would also be the same so far as the shape of the diagram is concerned. The pressure P_2 might, however, be maintained at a lower value, thus giving a greater temperature range.

The pressure rise ea within the cylinder would result directly from the opening of the admission valve and the admission of steam for the next cycle. But, if the working substance is to be returned to starting conditions as was done in the ideal case, its pressure must also be raised to P_1 and its temperature to a corresponding value. The pressure is raised in the case of condensing operation by means of the *boiler feed pump,* which picks up the condensed steam (condensate) and forces it into the boiler. The temperature of the working substance is raised by passing it through *feed-water heaters* or by heat absorbed directly from the heated water in the boiler.

When operating non-condensing the working substance exhausted during the last part of each cycle is really thrown away by allowing it to mix with the atmosphere, but an equivalent quantity of water is fed to the boiler by the boiler feed pump and takes the place of the material lost by exhaust to atmosphere. This method of operating does not approximate the ideal as closely as does the condensing method, but the discrepancy is not very great.

53. Losses in Real Installations. The diagram given in Fig. 33 was obtained by assuming the absence of certain practical losses and is considerably modified when real

apparatus is used. Thus the real engine, as shown in connection with Fig. 9, has *clearance* and operates with *compression* so that the clearance is filled with steam at a pressure indicated by the point a' in Fig. 34 when the admission valve opens.

There is also always some drop of pressure along the steam pipe so that the pressure at the engine is lower than at the boiler. Further, the admission valve can never be made to give such a large opening into the cylinder that there is not a measurable drop of pressure in flowing through it. As a result of these actions the highest pressure attained within the cylinder as indicated at point a in Fig. 34 is always lower than the boiler pressure P_1.

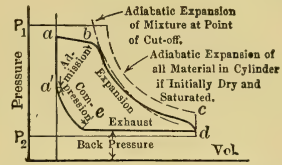


FIG. 34.—Theoretical and Real Indicator Diagrams.

As the piston of a real engine moves out it acquires a higher and higher velocity until it reaches a point near mid-stroke. The entering steam therefore must flow through the valve with increasing velocity if it is to follow up the piston and fill the cylinder, but this usually necessitates greater pressure drops as the piston moves out, so that the admission line generally slopes downward instead of being horizontal. There is also another phenomenon which causes this line to slope. The metal of cylinder, cylinder head and piston is in contact with comparatively low-temperature steam during the latter part of each cycle, and therefore acquires a lower temperature than that of the steam about to enter. Therefore, when the high-pressure steam enters the cylinder it gives up heat to the walls at a comparatively rapid rate, and, if initially dry saturated, this results in a great deal of condensation. Such condensation is called **initial condensation.**

As the steam condenses after flowing into the cylinder and forms water occupying a negligibly small volume,

it follows that steam must flow into the cylinder at a proportionately greater rate in order to fill the space vacated by the piston. But this results in an increased pressure drop and therefore would give a sloping admission line.

When the piston has finally been driven out as far as desirable by the action of high-pressure steam, the admission valve is closed, that is, **cut-off** occurs. This valve can not be closed suddenly; the closure is more or less gradual in all cases. As the opening becomes smaller it becomes increasingly more difficult for the steam to flow through and into the cylinder so that the pressure continues to drop at an increasing rate until the valve is finally closed. This gives the rounded cut-off shown at the point *b*.

The loss of pressure during admission is generally said to be due to **throttling** or **wire drawing**, these terms being intended to convey the idea that the steam has to squeeze its way through the inlet openings with corresponding loss of pressure.

When the cut-off has finally been completed, it leaves the end of the cylinder filled with a mixture of steam and water at steam temperature, and this mixture then expands as shown by the line *bc*. At the beginning of the expansion the steam generally has a higher temperature than that of the surrounding walls and it therefore continues to give heat to those walls. Were the expansion adiabatic it would follow the dot-dash line in the figure, but, as the steam must not only convert heat into work, but must also supply heat to the walls, it condenses more rapidly than in the ideal case and its pressure and volume changes follow some such law as that indicated by the upper part of the curve *bc*.

As expansion continues, the pressure and temperature of the steam drop until some point is reached at which the temperature has become equal to that of the walls. Further expansion with drop of pressure and temperature results in reducing the temperature of the steam below

that of the walls, and then the direction of heat transfer is reversed, the hot walls giving heat to the cooler steam at an increasingly rapid rate. This heat causes *re-evaporation* of some of the water formed before and thus tends to increase the volume occupied by the material in the cylinder, with the result that the lower part of the expansion curve *bc* approaches and generally crosses the curve which would have been attained by adiabatic expansion in non-conducting apparatus.

In many real engines the re-evaporation is so great that the steam is entirely dried and sometimes superheated before the exhaust valve opens.

The exhaust valves of steam engines are always opened before the piston reaches the end of the stroke, as it is found necessary to do this if excessive losses are not to occur due to the difficulty of forcing the large volume of low-pressure steam through the exhaust passages. When opened early enough, the steam flows out in such quantity before the end of the stroke that the " back pressure " during the return or exhaust stroke is only a pound or two above that of the space into which the engine is exhausting.

During all of the exhaust period, the steam is probably at a lower temperature than the walls to which it is exposed and re-evaporation probably continues in most cases until the closure of the exhaust valve. It seems probable that the steam retained in the cylinder after the closure of the exhaust valve is approximately dry, but little is really known regarding the quality of the clearance steam.

The rise of pressure during compression has two beneficial effects: It helps to bring the moving parts to rest gradually, and it raises the temperature of the clearance steam and of the walls of the clearance space to values nearer that of the entering steam.

Remembering that area on a *PV*-diagram represents work, it is easily seen that throttling losses and rounding of corners due to slow valve action (which cause a loss of

diagram area) result in a loss of work. The fact that condensation also causes a great loss is easily shown. A given quantity of steam entering the engine with its supply of heat can, in the ideal case, do a certain amount of work at the expense of that heat. In the real case part of the heat is stored in the walls during the early part of the cycle, so that it is not available for the doing of work and is removed from the walls and carried out into the exhaust as unutilized heat during the later part of the cycle. The phenomenon can be pictured by imagining the steam as dropping some of its heat into a pocket in the walls of the cylinder when entering the engine and then picking it up again and carrying it out when leaving, so that the next charge of steam will have to fill the pocket again.

The net result of condensation and re-evaporation is the obtaining of less work from a given quantity of steam than should be obtained, or the use of more steam than theoretically necessary for a given quantity of work. This effect is shown graphically by the two adiabatic expansion lines of Fig. 34.

The initial condensation in real engines which are supplied with saturated steam generally amounts to from 20 to 50 per cent of all the steam supplied, so that it is evident that anything which will prevent part or all of this loss should do much to improve the steam consumption of engines. This subject will be discussed in more detail in later paragraphs and various methods of decreasing losses from this source will be considered.

54. Clearance. The term clearance is used in a twofold sense; (*a*) to refer to **mechanical clearance** or the *linear distance* between the two nearest points of cylinder head and piston face when the piston is at the end of its stroke, and (*b*) to refer to **volumetric clearance** or the *volume* enclosed between the face of the valve, the cylinder head and the face of the piston when the latter is at the end of its stroke.

The former is generally given in inches and varies from a very small fraction of an inch in the best engines to an inch or more in cheap and in poorly designed engines. It is indicated by a in Fig. 35.

The volumetric clearance is expressed as a percentage of the piston displacement or volume swept through by the piston. It varies from 2 per cent or less in the best engines to as high as 15 per cent in the cheaper and less economical models. It is made up of the parts designated by c in Fig. 35.

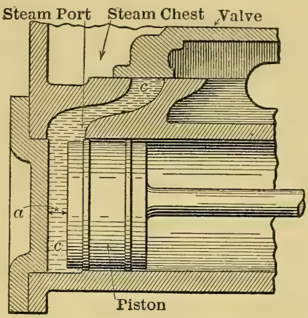


Fig. 35.—Mechanical and Volumetric Clearances.

55. Cushion Steam and Cylinder Feed. It is customary to imagine the steam operating within an engine cylinder to consist of two parts, the *cushion steam* and the *cylinder feed*. The former is that part of the total which is contained in the clearance space before the admission valve opens and serves to *cushion* the reciprocating parts of the engine. The cylinder feed is the steam which enters through the valve for each cycle.

If the same cycle is produced time after time so that all temperature effects are repeated at regular intervals and so that all events occur at the same points in successive cycles, the quantity of steam retained in the clearance volume will be the same for successive cycles. It is impossible to measure the quantity of this steam directly and indirect methods are therefore adopted for that purpose.

It is often assumed that the steam is dry and saturated when compression begins, as at the point e in Fig. 34. With this assumption, the weight of cushion steam can be determined by dividing the volume occupied, that is,

V_e, by the volume occupied by one pound of dry saturated steam at the same pressure. Thus,

$$\text{Cushion steam} = \frac{V_e}{\text{Sp.vol. at pressure } P_e} \text{ lbs.} \quad . \quad (29)$$

The weight of cylinder feed can be very accurately determined by condensing and weighing the steam leaving the engine in a given time and dividing by the number of cycles performed during the same period. It can also be determined by metering the steam entering the engine or by measuring the water fed to a boiler supplying only the engine in question. An approximate determination of the quantity of the cylinder feed can also be made directly from an indicator diagram by determining what is known as the **diagram water rate**. This will be considered in detail at a later point.

When cushion-steam and cylinder-feed have both been determined, the *weight* of steam contained in the cylinder between cut-off and release can be found by adding the two quantities. Thus,

$$W = W_f + W_K, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

in which

W = total weight of steam expanding in cylinder per cycle;
W_f = weight of cylinder feed per cycle; and
W_K = weight of cushion steam per cycle.

The *volume* which the mixture would occupy if dry and saturated at any given pressure can be determined by multiplying W the total weight by the specific volume for that particular pressure.

56. Determination of Initial Condensation. The loss due to initial condensation is so important that it is customary to determine the amount of this loss when studying engines. This can be done with fair accuracy by means of the indicator diagram.

To make such a study it is necessary to know the total weight of material in the engine cylinder at the point of cut-off. This weight may be determined by any of the methods just given. With the weight known, the volumes which this material should occupy at different pressures if dry and saturated can be determined by multiplying by the specific volumes at the various pressures. Plotting these points on a PV-diagram and connecting them will give a *saturation curve* for the material in the cylinder such as the curve shown in Fig. 36.

By drawing this curve on the indicator diagram obtained from the engine and then comparing distances such as ab and ac as explained in section 26 of Chapter III

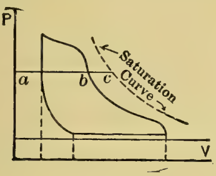


Fig. 36.

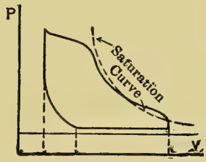


Fig. 37.

the quality of the steam within the cylinder at all pressures between cut-off and release can be determined. The weight of initial condensation (up to the point of cut-off) must be the total weight of water shown as existing within the cylinder at that point minus any water brought in by the steam if it was not dry when entering the engine.

Should the saturation curve cross the real expansion curve, as shown in Fig. 37, it indicates that the steam occupies volumes greater than the specific volumes toward the end of the expansion; the steam within the cylinder must therefore be superheated during this part of the cycle.

Many formulas have been devised for giving the quantity of initial condensation. They are all based upon the results of experiment and generally only give reliable

values for cases similar to those used in developing them. One formula of this sort which has been very widely tested and been found to give reliable results within its field of applicability is that devised by Robert C. H. Heck and explained in his books on the steam engine. The formula is

$$m=\frac{c}{\sqrt[3]{N}}\sqrt{\frac{s\theta}{pe}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

in which

m = the fraction representing initial condensation; for ordinary cases it is the fraction of the cylinder feed which is condensed during admission, but when compression is very high and when great weights of steam are retained in the clearance it is the fraction of all the material within the cylinder which exists in liquid form at the time of cut-off;

c = a coefficient, which varies between 0.25 and 0.30 with ordinary engine types. Its value is unknown for certain new types such as the Una-flow.

N = engine speed in revolutions per minute (R.P.M.);

s = a constant for any engine, equal to nominal surface in square feet divided by nominal volume in cubic feet. The nominal surface is the area of the inner walls and the ends of a cylinder with diameter equal to the internal diameter of the cylinder and with a length equal to the stroke of the engine. The nominal volume is the cubic contents of such a cylinder;

$s=\frac{12}{D}\left(2\frac{D}{S}+4\right)$ in which D and S represent diameter and stroke of engine in inches;

θ = a temperature function obtained from Table II as there indicated;

p = the absolute pressure in cylinder in pounds per square inch just after completion of cut-off;

e = cut-off ratio, that is, ratio of cylinder volume opened up by time cut-off has just been completed, to the total piston displacement.

TABLE II

FOR FINDING VALUES OF θ FOR USE IN HECK FORMULA

$\theta = k_1 - k_2$ when k_1 and k_2 are chosen from table for highest and lowest pressures existing in cylinder

p	k	p	k	p	k	p	k	p	k	p	k
1	175	15	210	50	269.5	90	321.5	160	389	230	441
2	179	20	220	55	277	100	332.5	170	397	240	447.5
3	183	25	229	60	284	110	343	180	405	250	454
4	186	30	238	65	291	120	353	190	413	260	460.5
6	191	35	246	70	297.5	130	362.5	200	420	270	467
8	196	40	254	75	304	140	371.5	210	427	280	473
10	200	45	262	80	310	150	380.5	220	434	290	479

57. Methods of Decreasing Cylinder Condensation. Before discussing methods of decreasing the loss due to cylinder condensation it will be well to consider what things may be expected to determine the extent of such loss. The condensation is due directly to the transfer of heat from one body to another at lower temperature, and anything which tends to increase the total amount of heat thus transferred will increase the total condensation.

It is therefore evident that the ratio of steam condensed to steam supplied will be greatest when:

(*a*) The time of contact is greatest;

(*b*) The ratio of surface exposed to volume enclosed is greatest, and

(*c*) The temperature difference is greatest.

The time of contact can be controlled to a certain extent by controlling the speed of the engine and, with other things equal, the higher the speed the lower should be the condensation.

The ratio of surface exposed to steam to the volume occupied by steam has a great influence on the amount of

condensation which occurs. The surface of the clearance space, including the interior surfaces of all ports or passages leading to the valves, seems to have the greatest influence, and the clearance space which is most nearly a short cylinder without connected passages may be expected to give the least initial condensation.

The size of the engine is also important in this connection. The area exposed does not increase as rapidly as does the volume inclosed when the diameter of a cylinder is increased, and therefore large cylinders give smaller ratio of surface to volume and therefore a smaller percentage of steam condensed. Large engines thus have a decided advantage over small engines.

The shape of the cylinder also has an effect. The longer the cylinder with respect to its diameter the more favorable its performance.

The point at which cut-off occurs is also intimately connected with the condensation loss. In a given cylinder with a given clearance the total condensation within the clearance space may be assumed practically constant if speed and temperature remain about the same. But if the cut-off is made later larger quantities of steam are admitted per stroke, and hence the fraction of the total cylinder feed which is condensed decreases.

The temperature differences depend on upper and lower pressures, that is, on the pressure range. The inner surfaces of the walls follow as rapidly as possible the temperature changes of the steam within them. Thus their average temperature is somewhere between the upper and lower temperatures of the steam. If now, with a given upper steam pressure and therefore temperature, the lower pressure be reduced, the average wall temperature also will be reduced, and therefore the differences between the temperature of the entering steam and the average temperature of the walls will be increased with a resulting increase in condensation loss.

The methods of decreasing this loss can now be considered. They are given below under separate heads with brief explanation when necessary.

(*a*) Clearance spaces should be properly designed so that the minimum surface is exposed.

(*b*) The proportions of cylinder (diameter and stroke) and the speed of the engine should be so chosen that the condensation loss is reduced to a minimum.

(*c*) The engine should be so proportioned that when delivering its rated power the cut-off occurs at such a point as to make the percentage of cylinder condensation the minimum consistent with other requirements.

(*d*) The cylinder should be surrounded by spaces filled with air or by materials which are poor conductors of heat so as to decrease loss by radiation, because all heat lost in this way must be supplied by the condensation of steam within the cylinder. Such metallic parts as cannot be "lagged" in this way should be polished because polished surfaces radiate less heat than dull surfaces under like conditions.

(*e*) The cylinder may be surrounded by a steam jacket, that is, a space filled with steam similar to that supplied the cylinder. The use of such a jacket sometimes results in a considerable saving and at other times in a great loss. The cylinder proportions, speed and pressure range seem to be the determining factors, and most long-stroke cylinders operating at low rotative speed and with small pressure ranges are jacketed.

(*f*) The engine may be compounded, that is, the expansion of the steam may be made to occur in two or more cylinders taking steam in series. This results in decreasing the pressure range in each of the cylinders and effects a decided saving under proper conditions. Compounding will be considered in detail in a later chapter.

(*g*) The engine may be supplied with superheated steam. If the steam is sufficiently superheated it can give up part or all of its superheat to heat the cylinder walls, and thus no

condensation need occur. Heat interchanges between metal and superheated steam also appear to be less rapid than is the case when the steam contains water, so that a saving results from this source also.

Tests made with saturated and with superheated steam indicate that from 7° to 10° of superheat are generally required to prevent 1 per cent of initial condensation. Results differ greatly with the character of the engine, with its economy on saturated steam, with its valve gear, etc. Superheats of from 25° to 50° can generally be used with any well-designed engine, but higher temperatures usually require specially constructed engines. With proper cylinder and valve construction the maximum permissible temperature is set by the lubricating oil. At the present time the maximum steam temperature is thus limited to 600° F. or less.

(*h*) The engine may be built to operate on the Una-flow cycle described later. In this case a very ingenious modification of the engine results in decreasing the temperature differences between walls and steam, with resulting diminution of heat transfer.

58. Classification of Steam Engines. Steam engines, are classified on many different systems, the one used in any particular case being determined largely by circumstances. The principal methods of classification are indicated in the following schedule:

Classification of Steam Engines

On the basis of rotative speed.

(*a*) Low speed; (*b*) Medium speed; (*c*) High speed.

On the basis of ratio of stroke to diameter.

(*a*) Long stroke; (*b*) Short stroke.

On the basis of valve gear.

A. Slide valve; (*a*) D-slide valve; (*b*) Balanced slide valve; (*c*) Multiported slide valve; (*d*) Piston valve.

B. Corliss valve; (*a*) Drop cut-off; (*b*) Positively operated.

C. Poppet valve.

On the basis of position of longitudinal axis.

(*a*) Vertical; (*b*) Inclined; (*c*) Horizontal.

On the basis of number of cylinders in which steam expands.

A. Single expansion or simple engine.

B. Multi-expansion engine. (*a*) Compound expansion; (*b*) Triple expansion; (*c*) Quadruple expansion.

On the basis of cylinder arrangement.

(*a*) Single cylinder; (*b*) Tandem compound; (*c*) Cross compound; (*d*) Duplex.

On the basis of use.

(*a*) Stationary engines; (*b*) Portable engines; (*c*) Locomotive engines; (*d*) Marine engines; (*e*) Hoisting engines.

59. Rotative Speeds and Piston Speeds. High-speed engines operate at a comparatively high rotative speed and are characterized by a short stroke in comparison with the diameter of the cylinder, the stroke generally being equal to, or less than, the diameter. The piston speed, by which is meant the feet travelled by the piston per minute, generally falls between 500 and 700.

It is not considered advisible to allow piston speeds of stationary steam engines to exceed about 750 feet per minute for ordinary constructions and the great majority of engines give much lower values. The piston speed will obviously be given by the formula

$$S=2LN, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

in which

S = piston speed in feet per minute;
L = stroke in feet; and
N = revolutions per minute,

and it is evident from this formula that as the rotative speed is increased the piston speed will increase unless the length of stroke is proportionately decreased. As a result, high-speed engines have short strokes in com-

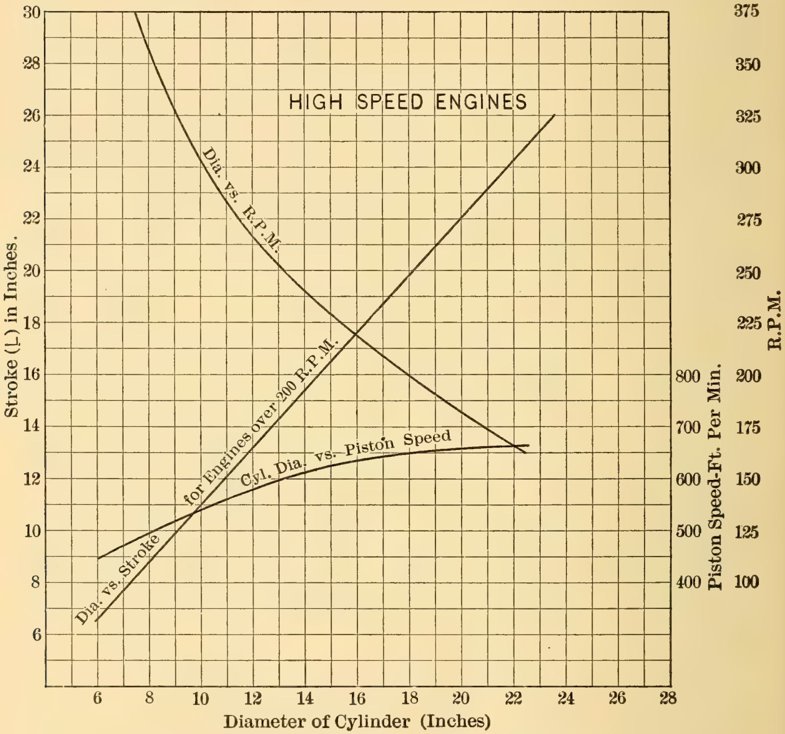


FIG. 38.—Proportions of High Speed Engines.

parison with their cylinder diameters and slow-speed engines have long strokes.

The characteristic relations between cylinder diameter and stroke, rotative speed and piston speeds of high-speed engines are given in Fig. 38.

High-speed engines are generally fitted with some

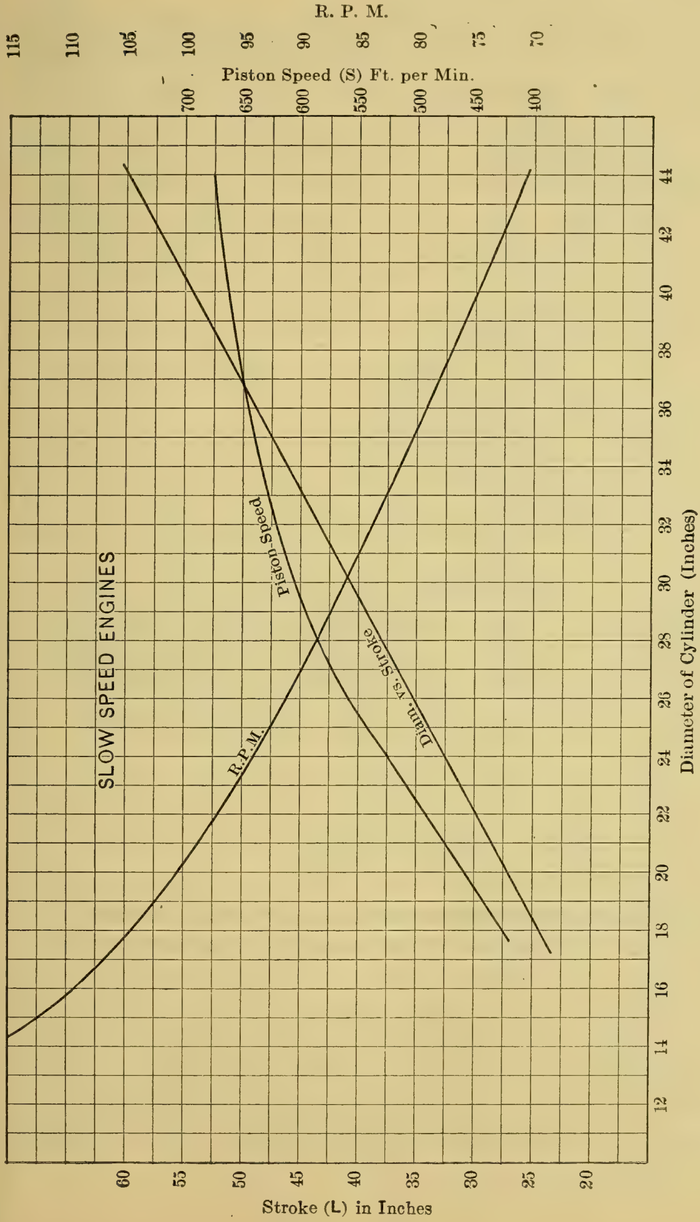

R. P. M.
115 110 105 100 95 90 85 80 75 70
Piston Speed (S) Ft. per Min.
700 650 600 550 500 450 400
SLOW SPEED ENGINES
Piston Speed
Diam. vs. Stroke
R.P.M.
12 14 16 18 20 22 24 26 28 30 32 34 36 38 40 42 44
Diameter of Cylinder (Inches)
60 55 50 45 40 35 30 25 20
Stroke (L) in Inches

Fig. 39.

form of balanced slide valve, and are controlled by what is called a *shaft governor*. They are very compact, having small weight and occupying small space in comparison with the power developed.

Slow-speed engines are, in general, the most economical and are characterized by low rotative speed, long stroke, and elaborate valve gear. The weight per horse-power is high, and they generally occupy a great deal of space. The *Corliss engine* is the best known and most widely built engine of this type in this country.

The characteristic relations of cylinder diameter to stroke, rotative speed and piston speed for slow-speed engines are given in Fig. 39.

Medium-speed engines generally operate at rotative speeds between 150 and 250 R.P.M. They are generally fitted with the better forms of multiported and balanced slide valves, with poppet valves, or with a positively operated Corliss type of valve.

60. The Simple D-slide Valve Engine. The simplest and cheapest type of reciprocating steam engine manufactured is shown in part section in Fig. 40 with the principal parts labelled. The cylinder, piston, steam chest and valve are sectioned in order to show the internal construction.

This engine, like most steam engines, is double acting, that is, a cycle is produced on each side of the piston during every revolution. Steam is admitted and expanded on one side of the piston while steam is being exhausted on the other side. The control of admission and exhaust is effected by the slide valve and will be considered in detail in later sections.

The mechanical energy made available by the steam operating in an engine cylinder is not developed at a uniform or constant rate, but fluctuates, during each revolution, above and below the amount required to overcome the constant resistance at the shaft due to the work the engine

is doing. If no provision were made to prevent it, this would result in a very variable rate of rotation during each revolution. When the energy made available was in excess of the demand it would be used in accelerating the moving parts of the engine and the speed of the latter would increase. The reverse would occur when the supply did not equal the demand.

The fly-wheel is used to prevent violent fluctuations

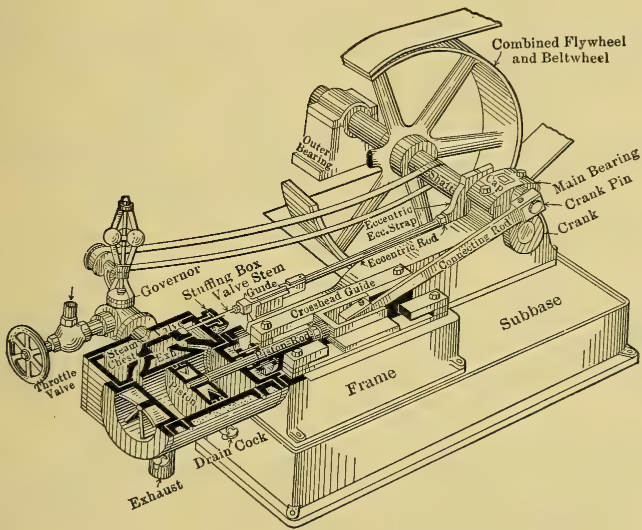


FIG. 40.—Simple *D*-slide Valve Engine.

of this kind. It is made with a comparatively heavy rim and a great deal of energy must be supplied to accelerate it to any appreciable extent in a short time. Similarly it can give out a great deal of energy when slowing down. The fly-wheel therefore serves as a sort of reservoir in which excess energy can be stored temporarily and from which it can later be withdrawn when a deficiency exists. The fly-wheel thus acts as a damper to variation of rotative

speed during each revolution, minimizing but not entirely eliminating such variation. It may also serve as a belt wheel, as shown in the illustration.

The governor controls the steam supply to the cylinder in such a way that enough heat will be supplied to make available the power demanded at the shaft. Were more supplied the excess would be absorbed by the moving parts and the engine speed would increase, were less supplied the engine speed would decrease.

61. Engine Nomenclature. The meanings of several terms used in describing engines are not self-evident, their

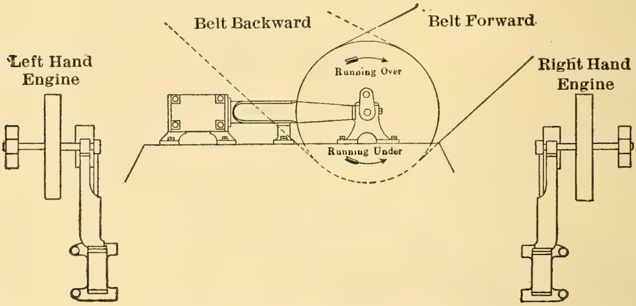


FIG. 41.—Engine Nomenclature.

definitions depending merely on accepted usage. Some of these terms and their meanings are illustrated in Fig. 41.

The crank end of a horizontal engine is called the **front** of the engine, so that the cylinder head nearest the crank is called the front head and the stroke of the piston toward the crank is known as the **forward stroke**. The forward stroke of the piston is also spoken of as the **outstroke**, particularly in connection with single acting engines. The stroke away from the crank is correspondingly designated as the return or the **instroke**.

62. Principal Parts of Engines. The parts of engines may be roughly divided into *stationary* and *moving*, such as frame, cylinder, cylinder and valve chest covers, etc., which

are stationary, and piston, piston rod, crosshead, fly-wheel, etc., which are all moving parts when the engine is in operation. The moving parts are often divided into *reciprocating* and *rotating* parts. Thus the piston and all connected parts through and including the crosshead, and the valve and many connected parts in the case of slide valve engines, all reciprocate when the engine is in operation. The shaft, fly-wheel, eccentric sheaves and governor constitute the principal rotating parts.

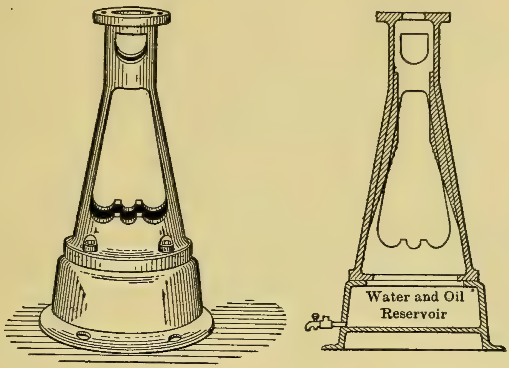


FIG. 42.—Frame for Small Vertical Engine.

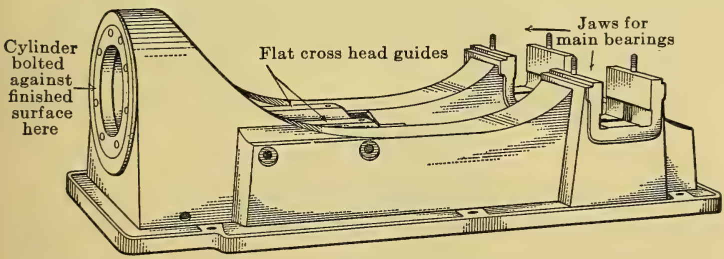


FIG. 43.—Frame for Medium Speed Center Crank Engine.

Some engines also have *oscillating* parts, such as the valves in Corliss types, which rock back and forth in the arc of a circle, and the rocker arms in various forms of valve gear, these arms rocking through a short arc about a fixed pin near one end.

The principal parts and their functions are briefly considered in the following paragraphs:

(*a*) **The Frame.** The frame of the engine, sometimes known as the *bed*, serves to support the other parts, to tie

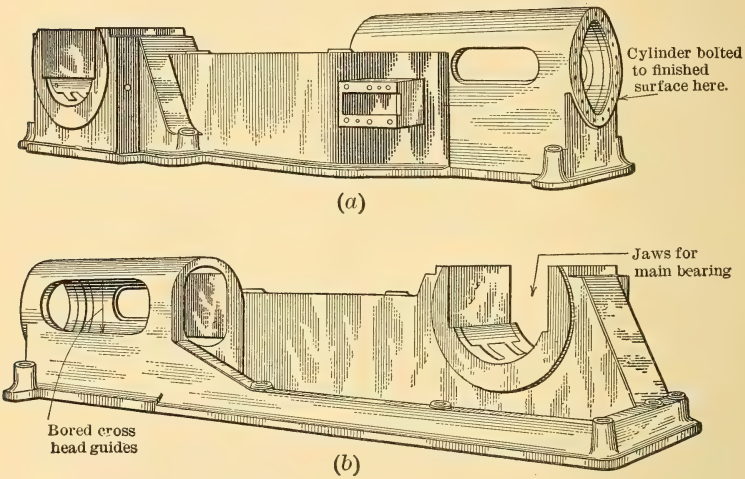


Fig. 44.—Frame for Slow Speed Engine of Corliss Type; Side or Overhung Crank.

them together in their proper relations and to fasten the whole structure to whatever foundation is used. The cross-head guides and the seats for the main bearings are incorporated in the frame.

The frame is commonly made of cast iron in the form of a hollow box which is properly ribbed to give the necessary stiffness.

Examples of frames are shown in Figs. 42, 43 and 44.

(*b*) **The Cylinder and Steam Chest.** The cylinder and steam chest are generally incorporated in the same casting, and surfaces of covered cavities in this casting are finished

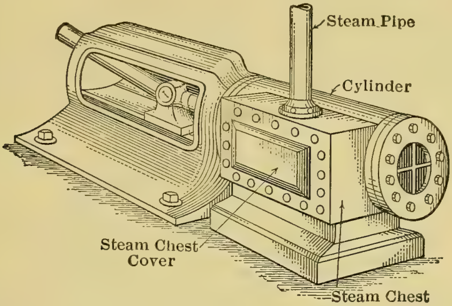


FIG. 45.

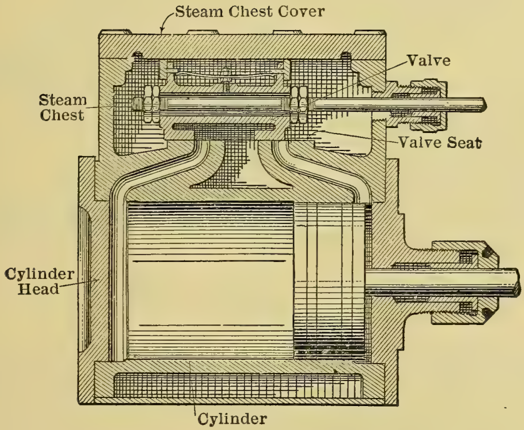


FIG. 46.

to form the cylinder in which the piston operates and the seat or seats upon which valves rest and move.

The cylinder may be *single walled* with flanges on the end to receive the cylinder head, as illustrated in Fig. 40

(plain *D*-slide valve), in which case a thin sheet-metal jacket is fastened around it and the space between filled with heat-insulating material. Or, the cylinder may be cast with *double walls*, the space between the two being used as an air jacket or as a steam jacket.

Some cylinders are fitted with a *liner*, which is a plain cylinder pressed into place within the cylinder casting and forming the bore of the working cylinder. This prac-

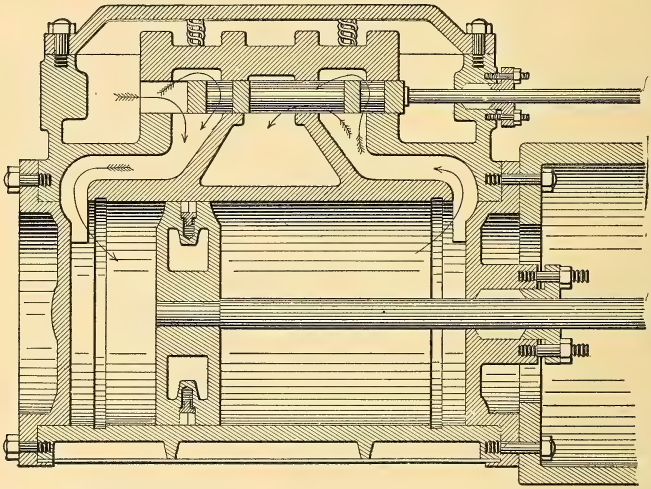


Fig. 47.—Section of Atlas Medium Speed Engine, Showing Balanced Slide Valve.

tice is common on the larger types, the liner being used so that when wear has occurred it can be replaced cheaply, instead of it being necessary to rebore or even replace the main casting.

Examples of cylinder construction are shown in Figs. 45, 46, 47, 48, 49 and 50.

(*c*) **The Piston.** The function of the piston is two-fold. It must prevent the leakage of steam by it from one end of the cylinder to the other, and it must receive the

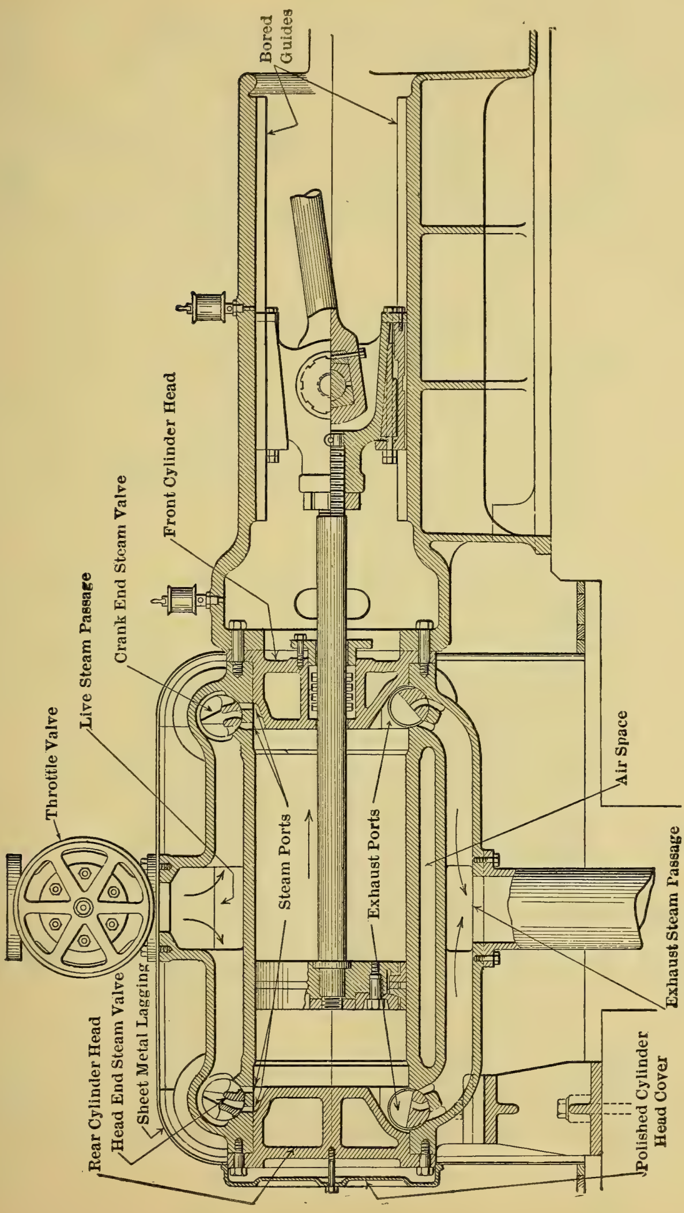


FIG. 48.—Section of Corliss Cylinder and Guides, and Half-section of Crosshead.

pressures exerted by the steam and transmit them to the other parts of the mechanism as it moves.

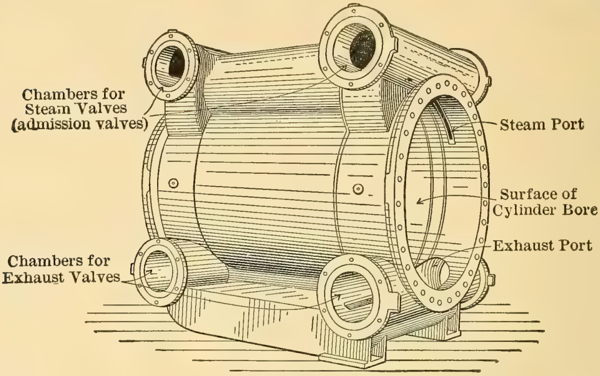


FIG. 49.—Corliss Cylinder Casting.

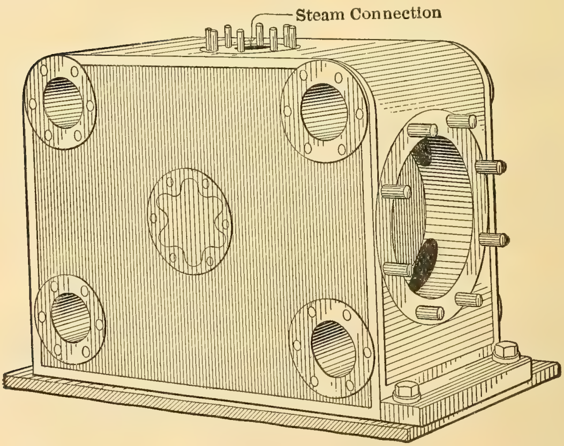


FIG. 50.—Corliss Cylinder with Lagging in Place.

Leakage of steam is prevented by the use of piston rings, which are metal rings fitted into grooves in the circular surface of the piston and pressed out against the cylinder

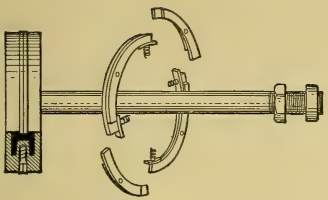


FIG. 51.

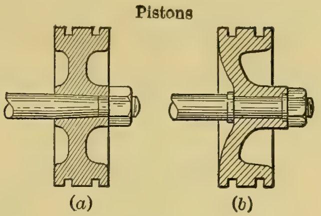


FIG. 52.

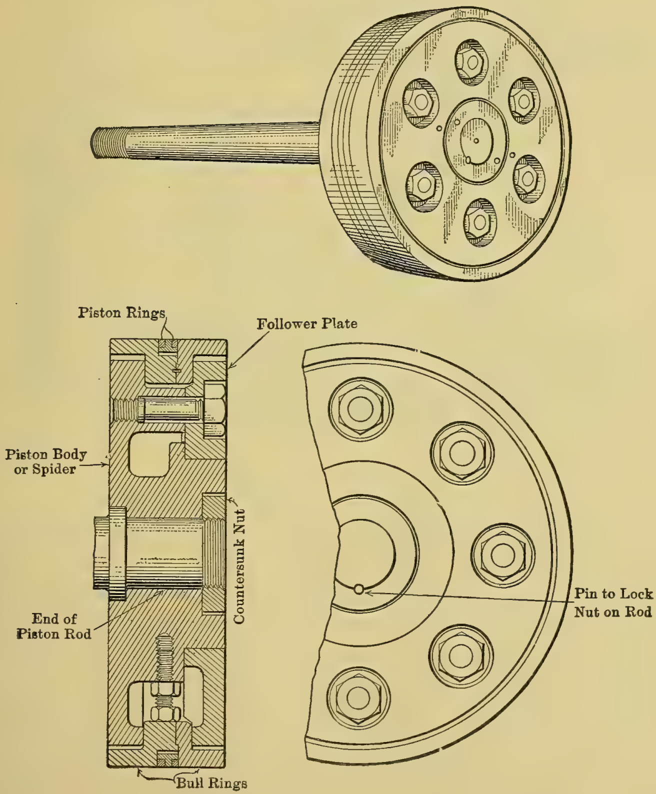


FIG. 53.—Built-up Piston Used in Large Engines.

walls by spring action. They may be made of one piece of metal turned into a ring of slightly larger diameter than the cylinder, cut through and sprung into place, or they may be made in pieces as shown in Fig. 51, and pressed out against the wall by small helical or leaf springs.

The piston itself may consist of a solid disk of metal fitted with a hub and a short cylindrical part with grooves for the rings, as shown in Fig. 52 (*a*) and (*b*), or it may be an elaborate built-up structure, as shown in Fig. 53.

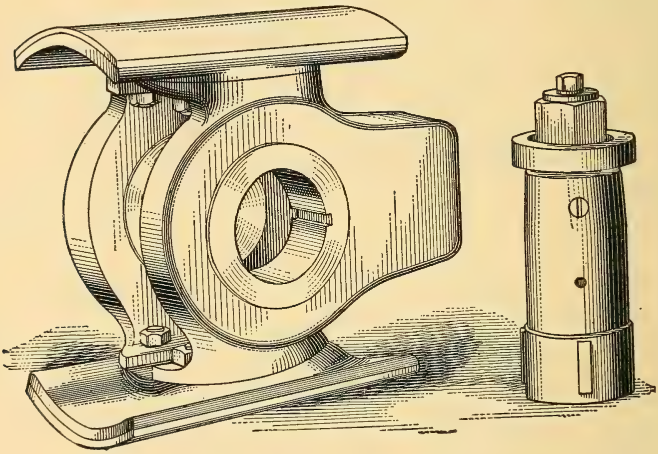
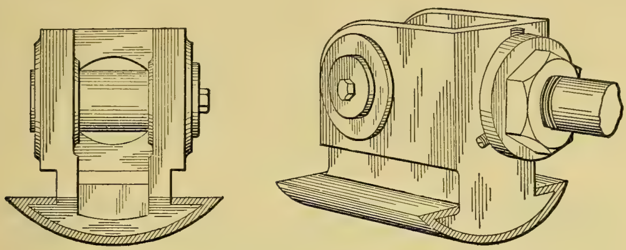


Fig. 54.—Crosshead and Pin.

(*d*) **The Piston Rod.** The piston rod is a plain cylindrical steel rod fitted with such shoulders and threads at the ends as are necessary for the fastening of the piston and the crosshead respectively. Examples of such fastenings are given in Figs. 51, 52, 53 and 55.

In large horizontal engines the piston rod sometimes extends through the piston and rear cylinder head, and the rear end is then supported by a small auxiliary crosshead. The extension of the rod is known as the **tail rod** and the auxiliary crosshead as the tail rod crosshead.

Such constructions are used when the weight of the piston is so great that it would cause serious cylinder wear if not supported more perfectly than is possible with the ordinary overhung arrangement.



Fig, 55.—Single Slipper Crosshead.

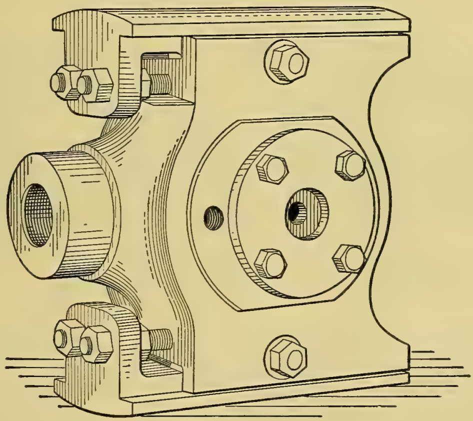


Fig. 56.—Crosshead with Adjustable Slippers.

(*e*) **The Crosshead and Guides.** The crosshead and guides are used for the purpose of supporting the piston and its rod and guiding them in a straight line. The crosshead also serves to connect the piston rod and the connecting rod through which the forces are transmitted to the crank pin.

Crossheads are generally cast in the form of imperfectly shaped boxes and carry slippers which are faced with anti-friction metal where they come in contact with the guides. The slippers may be flat and operate on planed guides, as shown in Fig. 40, or they may be turned and operate in bored guides, as shown in Figs. 48, 54, 55 and 56. Provision is generally, though not always, made for taking up wear of guides and slippers by setting the slippers further out from the body of the casting.

With the type shown in Figs. 40 and 54 this can be

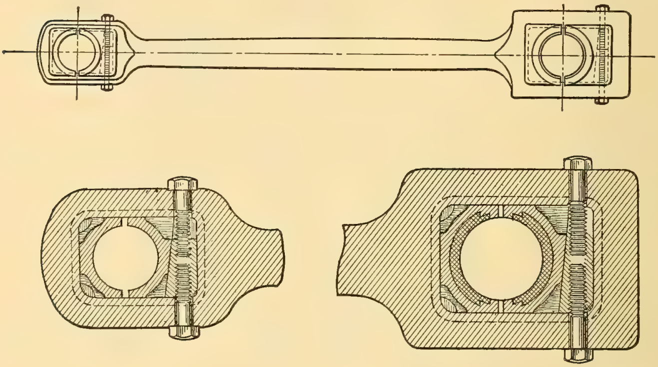


FIG. 57.—Solid End Connecting Rod; for Overhung Cranks only.

done by the insertion of thin sheets of metal or paper (known as shims) between the body of the crosshead and the slippers. In the type shown in Fig. 56 the slippers are finished with inclined surfaces where they come in contact with the main casting, and the adjustment is made by wedging the slippers apart by the use of the adjusting bolts shown.

The wrist-pin end of the connecting rod enters the crosshead casting and is held in place by means of the wrist pin, about which it oscillates when the engine is in operation.

(*f*) **The Connecting Rod.** This rod connects the reciprocating crosshead with the rotating crank shaft and transmits the forces from one to the other. It consists of a body or shank and two ends or heads. The ends may

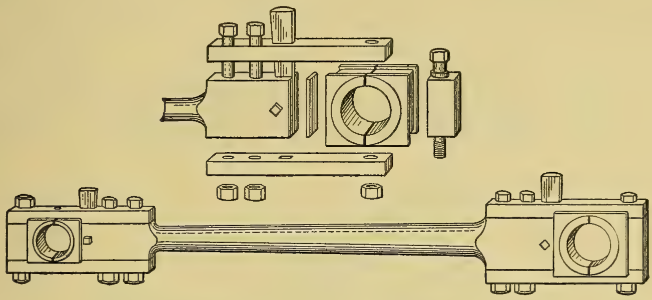


FIG. 58.—Connecting Rod with Bolted Strap Ends; May be Used with Center or Side Crank Constructions.

be "closed" or "solid" as shown in Fig. 57; they may be made with a strap bolted in place as shown in Fig. 58; or the crank-pin end may be made of two half boxes bolted together to form a "marine end" as shown in Fig. 59.

The ends are always made adjustable so that wear

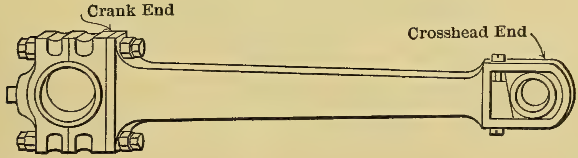


FIG. 59.—Marine End Connecting Rod.

can be taken up, thus preventing noisy operation due to hammering between the ends and the pins at times when the direction in which forces act is reversed. With solid and strap types this adjustment is generally made by means of wedges similar to those shown in Figs. 57 and 58.

With the marine type shims are used between the two halves, and the diameter of the hole formed by the latter is decreased by the removal of shims of the required thickness.

(*g*) **The Shaft.** The crank shaft itself is generally made

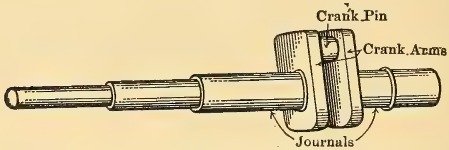


Fig 60.—Crank Shaft, Center Crank.

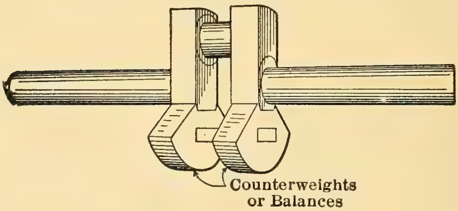


Fig. 61.—Center Crank.

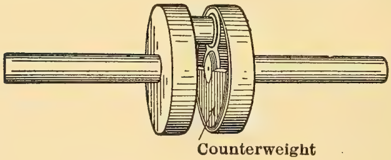


Fig. 62.—Center Crank.

of steel, but the counterbalances are often of cast iron. It may be one forging throughout or may be built up by shrinking the various parts together. Multicrank shafts of large size are generally of built-up construction, the crank pins being shrunk into the crank arms and the latter shrunk on to the pieces of shaft.

The counterbalance weights are used to balance the

centrifugal effect of the crank pin, part of the crank arms and part of the connecting rod, all of which rotate off center. In some engines part of the unbalanced effect of the reciprocating parts is also imperfectly balanced by these counterbalances.

Various types of shafts are shown in Figs. 60, 61, 62, 63, and 64.

(*h*) **Bearings.** Bearings are distinguished as *main* and as *outboard* bearings. Main bearings are those carried by

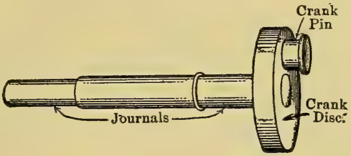


Fig. 63.—Crank Shaft and Disc, Overhung Crank.

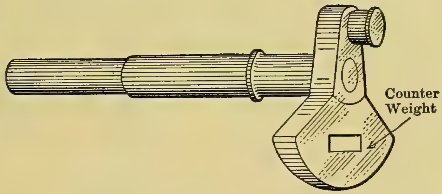


Fig. 64.—Overhung Crank.

the frame of the engine and outboard bearings are carried by separate pedestals or by pedestals fastened to a plate which is in turn fastened to the frame. Center-crank engines have two main bearings, and side-crank engines have only one, the other end of the shaft being supported by an outboard bearing.

The bearings of steam engines are generally formed of babbitt-lined boxes carried within jaws machined in a frame, or in a separate pedestal, and held in place by a bearing cap. The boxes are made in two, three or four parts to allow for adjustment to compensate for wear and

to give a certain degree of flexibility. Adjustment for wear is either made by means of wedges or by means of screws which force the various parts of the boxes toward the shaft. An example of a three-part bearing with screw adjustment as used with large side-crank engines is shown in Fig. 65. The parts of a three-part bearing with wedge adjustment are shown in Fig. 66.

Bearings are often lubricated by rings or chains, and they are then known as ring- or as chain-oiling bearings.

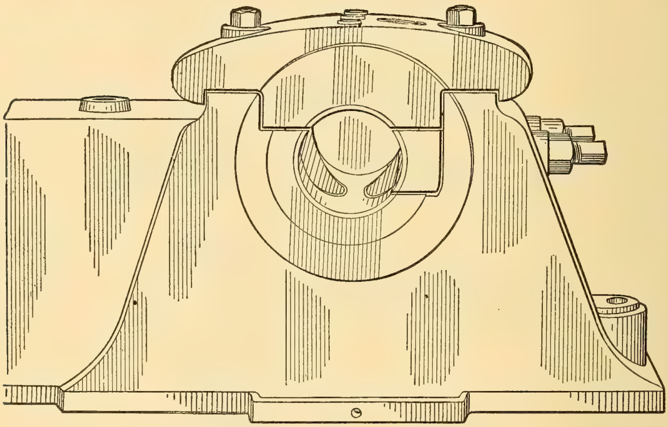


Fig. 65.—Three Part Main Bearing with Screw Adjustment.

In the ring-oiling bearing one or more metal rings of large diameter hang loosely on the shaft within the bearing and dip into an oil reservoir below the shaft. Rotation of the shaft causes the rings hanging on it to rotate and they carry oil up from the reservoir and spill it out over the shaft within the bearing. Chain bearings are similar except that chains are substituted for rings.

(*i*) **Fly-wheels.** The function of the fly-wheel has already been considered and need not be discussed further. The wheel is constructed with a heavy rim joined to a hub

by six or eight arms. In the smaller sizes the wheel may be cast in one piece, but best practice calls for a split hub in that case to partly equalize certain casting strains which result from unequal thicknesses of metal in different parts of the wheel. Large wheels are cast in two or more parts both for the purpose of partly avoiding casting strains and for the purpose of facilitating handling and shipping.

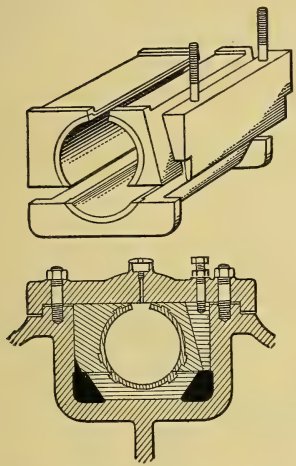


Fig. 66.—Three Part Bearing Showing Wedge Adjustment.

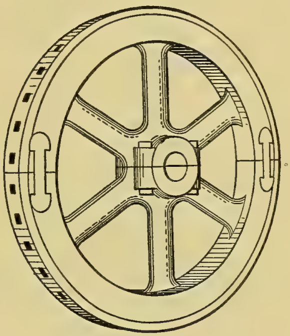


Fig. 67.

A two-part wheel with the rim sections joined by prisoner links shrunk in place and the hub fastened with bolts is shown in Fig. 67.

PROBLEMS

1. A given engine has a piston displacement of 3 cu.ft. and a clearance volume of 3%. Compression begins when 85% of the exhaust stroke has been completed and the pressure within the cylinder at that time is 16 lbs. per square inch absolute. Determine the weight of the cushion steam on the assumption that this steam is dry and saturated at the beginning of compression.

2. Assume the engine described in Prob. 1 to cut-off at $\frac{1}{4}$ stroke and with a pressure inside the cylinder equal to 115 lbs. per square inch absolute. Find the weight of cylinder feed if the quality of the material in the cylinder at the time of cut-off is 75%.

3. Find the piston speed of an engine with a stroke of 2 ft. and a rotative speed of 150 R.P.M.

4. Show by means of Heck's formula that initial condensation increases with pressure range.

CHAPTER VIII

THE INDICATOR DIAGRAM AND DERIVED VALUES

63. The Indicator. The ideal steam engine cycle was described in Chapter IV, and the sort of diagram which would be obtained from a real engine was shown in Chapter

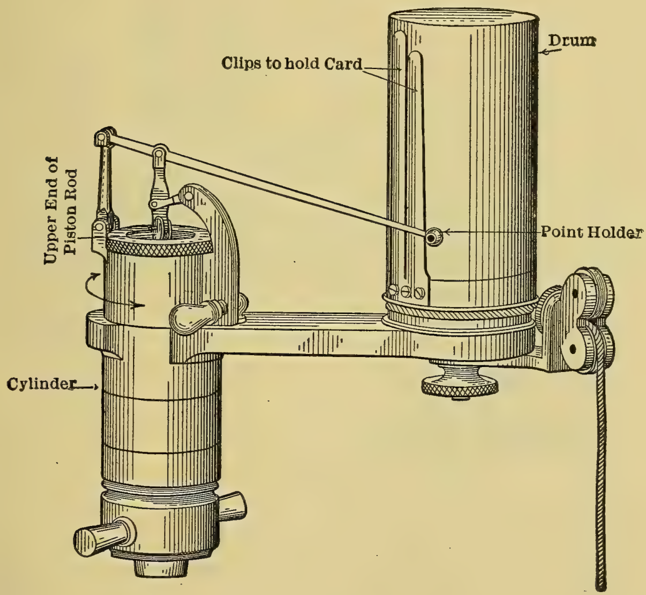


FIG. 68.

VII; but the means by which such diagrams are obtained from operating engines was not given.

Indicator diagrams showing the pressure and volume changes experienced by steam in the cylinders of real

engines are obtained by means of an instrument known as an **indicator**. The operation of obtaining such diagrams is known as *indicating the engine.*

An external view of one form of indicator is shown in Fig. 68 and a section through the instrument is given in Fig. 69. The method of connecting an indicator to the

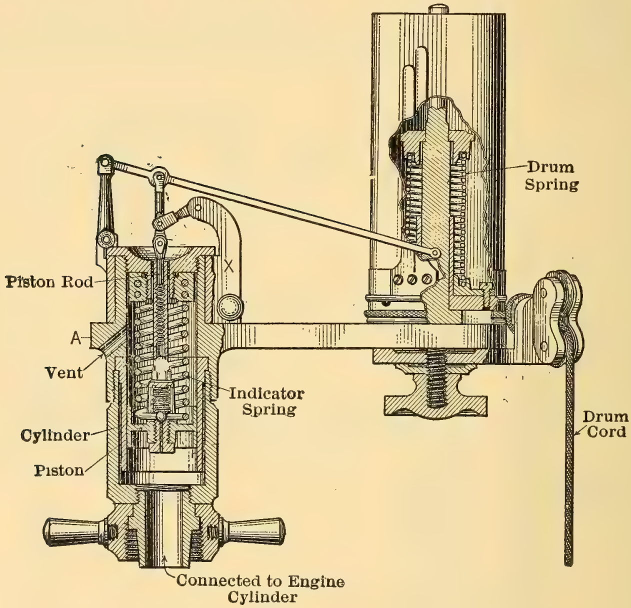


FIG. 69.

cylinder of a steam engine and one method used for driving it are illustrated in Fig. 70.

The indicator is intended to draw a diagram showing corresponding pressures and volumes within the engine cylinder and must, therefore, contain one part which will move *in proportion to pressure* variations and another which will move *in proportion to volume* changes. The one may

be called the pressure-measuring and the other the volume-measuring device.

The pressure-measuring device generally consists of a piston, such as shown in the figure, working with minimum friction in a small cylinder and fitted with a spring which will resist what may be called outward motion (upward in the figure). The cylinder containing this piston is coupled to a short pipe connected with the clearance space of the engine and, whenever the indicator cock in this connection is open, the steam acting on the engine piston

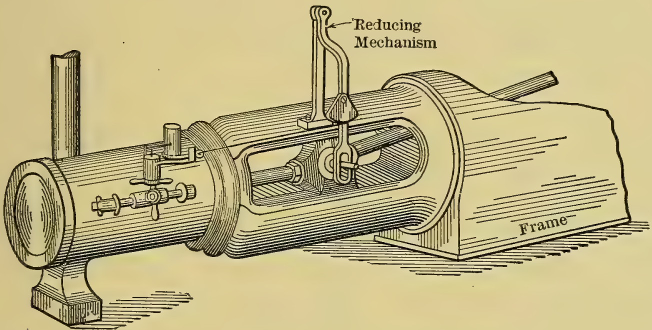


Fig. 70.—Method of Attaching and Operating an Indicator.

will also act on the indicator piston. Steam of any given pressure will drive the indicator piston out against the action of the spring until the pressure exerted by the spring is equal to that exerted on the face of the piston. The indicator piston will thus move out different distances for different pressures, and, through the piston rod and pencil mechanism, will move the pencil point to various heights corresponding to different steam pressures. The pencil mechanism is so arranged that the point traces a straight vertical line on the drum as the indicator piston moves in and out.

Springs are made to certain definite scales, thus there

are, for instance, 10-lb., 25-lb., 50-lb. and 100-lb. springs. The number which is known as the **scale of the spring** designates the steam pressure in pounds per square inch which is required to move the pencil point 1 inch against the action of such a spring. With a 100-lb. spring in the indicator, a steam pressure of 50 pounds per square inch acting on the indicator piston would drive the pencil up a distance of half an inch, a pressure of 100 pounds per square inch would give 1 inch of motion and so forth.

The volume-measuring device is of an inferential kind. It simply indicates the position attained by the engine piston at the time when a given steam pressure existed in the cylinder and the volume occupied by the steam can be calculated from piston position and cylinder dimensions. The position of the piston is indicated by connecting the cord wound around the drum to some part of the engine which is rigidly connected to the piston. The crosshead is commonly used for this purpose and, since the motion of this member is generally much greater than the circumference of the drum, it is necessary to use a **reducing mechanism** of some sort. This mechanism must be very accurate, so that it moves the drum as nearly as possible in proportion to the motion of the engine piston.

The pencil point moves up and down as the pressure within the cylinder varies, and the drum rotates under the point in proportion to the motion of the engine piston, so that the combination of the two motions brings the pencil point to successive positions on the drum which indicate successive corresponding values of steam pressure and piston position. By mounting a piece of paper, known as a card, on the drum and pressing the pencil point upon this paper, the successive positions occupied by the pencil point will be recorded in the form of a series of curves and straight lines.

If the drum is rotated with the lower side of the indicator piston connected to atmosphere, the pencil will trace

a horizontal line. This is known as the **atmospheric line** and is used as a reference for locating the pressure scale. If the indicator cylinder is then connected with the engine cylinder and the drum is rotated by the reducing mechanism, a diagram similar to that of Fig. 71 will be drawn upon the card. The atmospheric line indicates the height assumed by the pencil when atmospheric pressure acts on the piston and, knowing the value of the existing atmospheric pressure (barometer reading) and the scale of the spring, a line at a height representing zero pressure can be drawn on the card. This line is indicated in Fig. 72.

The length of the card between the lines *a* and *b* is proportional to the length of the engine stroke and there-

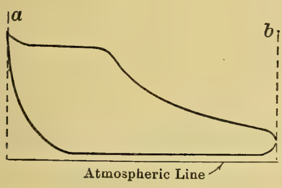


FIG. 71.

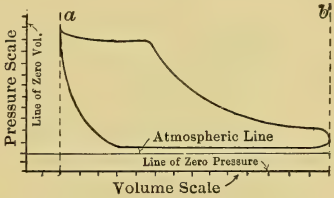


FIG. 72.

fore to the piston displacement, that is, to the *volume* swept through by the piston. Knowing the clearance volume of the engine as a percentage or fraction of the piston displacement, this fraction of the length of the diagram can be laid off from the end of the diagram to give a line of zero volume. This line is also indicated in Fig. 72.

With the line of zero pressure and the line of zero volume drawn in, all values of steam pressure and volume occupied by steam can be read directly from the diagram, and it thus forms a picture of what occurs within the real engine cylinder.

The indicator diagram is used for a number of purposes, the more important being:

(1) The determination of the energy made available within the cylinder, that is, the indicated horse-power, I.h.p.

(2) The determination of the amount of initial condensation and of heat interchanges between walls and cylinder.

(3) The determination of what is known as the diagram water rate.

(4) The study of the operation and timing of valves. The second one of these uses has already been considered in Chapter VII, the others are treated in succeeding sections.

64. Determination of I.h.p. The lines of the indicator diagram show by their height the pressures or forces acting on the engine piston as it moves. But *the product of force by distance is equal to work* and these lines can be used therefore for determining the net work done by the steam upon the piston.

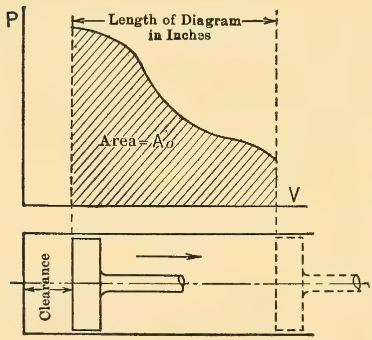


Fig. 73.—Positive Work Area.

In Fig. 73 is shown the upper part of the diagram, the curved lines representing the successive pressures in pounds per square inch which acted on the left face of the piston while it moved outward. If the *average pressure* could be determined and multiplied by the area of the piston face, this product would be the average total force acting on the piston. Multiplying this by the distance traveled would give the work done by the steam upon the piston. Expressed in the form of an equation,

$$E_0 = p_0 \times a \times L \text{ ft.-lbs.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

in which

E_0 = work done upon piston by steam during outstroke;

p_0 = mean pressure (in pounds square inch) acting on piston during outstroke;

a = area of piston face in square inches; and
L = stroke of piston in feet.

For the instroke shown in Fig. 74, the work done by the piston on the steam is given by the similar expression:

$$E_i = p_i \times a \times L \text{ ft.-lbs.}, \quad . \quad . \quad . \quad . \quad (34)$$

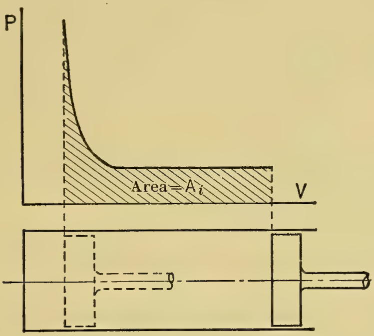


FIG. 74.—Negative Work Area.

in which E_i and p_i represent work done and mean pressure respectively.

The *net work* done by the steam upon the piston per cycle is then,

$$E_{\text{cycle}} = E_0 - E_i = (p_0 - p_i)aL \text{ ft.-lbs.} \quad . \quad . \quad . \quad (35)$$

The values of p_0 and p_i can be found directly from the diagram by dividing the areas A_0 and A_i respectively by the length l and then multiplying by the scale of the spring, giving

$$p_0 = \frac{A_0}{l} \times \text{scale of spring},$$

and

$$p_i = \frac{A_i}{l} \times \text{scale of spring},$$

so that,

$$p_0 - p_t = p = \frac{A_0 - A_t}{l} \times \text{scale of spring} \quad . \quad . \quad . \quad (36)$$

$$= \frac{\text{area of diagram}}{l} \times \text{scale of spring.} \quad . \quad (37)$$

The value of p evidently can be determined very simply from the indicator diagram, and the work per cycle can be found when p is known by substituting in the following equation, obtained by putting p for $p_0 - p_t$ in Eq. (35),

$$E_{\text{cycle}} = p \times a \times L \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad (38)$$

The pressure p is known as the **mean effective pressure** and is often represented by M.E.P.

If n cycles are produced per minute, the *net work* done by the steam upon the piston *per minute* will be

$$E_{\min} = p \times a \times L \times n, \quad . \quad . \quad . \quad . \quad . \quad (39)$$

which is generally rearranged to read,

$$E_{\min} = pLan, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

in which form the group of letters forming the right-hand member is easily remembered.

Since 33,000 foot-pounds per minute are equivalent to one horse-power, it follows that the power made available as shown by the indicator diagram, that is, the **indicated horse-power**, must be,

$$\text{I.h.p.} = \frac{pLan}{33{,}000}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

in which

p = mean effective pressure in pounds per square inch;
L = stroke of piston in feet;
a = area of piston in square inches;
$= (\text{diam. cyl. in inches})^2 \times \pi/4 = .7854d^2$; and
n = number of cycles per minute.

If an engine cylinder takes steam on one side of the piston only, that is, if the cylinder is *single acting*, the number of cycles produced per minute is equal to the number of revolutions per minute, but it should be noted that for other arrangements this is not necessarily true. In the case of *double-acting* engines which receive steam at both ends of the cylinder, the number of cycles produced is equal to twice the number of revolutions.

It should also be noted that the symbol a represents the area of the piston face upon which the steam acts. If a piston rod extend from the face of the piston to and through the cylinder head (as is always the case at the crank end of double-acting cylinders), the area of the piston rod must be subtracted from that of the piston to obtain the area on which the steam really acts. When a tail rod is used, a correction must be made for each side of the piston.

In the case of double-acting engines the indicated horse-power may be determined in two ways: It may be figured separately for the two ends of the cylinder, or the values for the area and pressure may be averaged for the two ends and the value of n chosen equal to twice the revolutions per minute. The former is generally the more accurate method.

It will have been observed that the area of the indicator diagram must be determined before the mean effective pressure can be found. This area is generally measured by means of an instrument known as a **planimeter,** and this is the most accurate method. It occasionally happens, however, that a planimeter is not available when the value of the indicated horse-power is desired. Under such circumstances an approximate determination of the area of the indicator diagram can be made by the method of ordinates.

For this purpose the length of the diagram is divided into an equal number of parts, usually ten, as shown in

Fig. 75 and vertical lines y_1, y_2, y_3, etc., are drawn at the center of each of the parts into which the diagram has been divided. The mean ordinate or height is then found from the equation,

$$y_m = \frac{y_1 + y_2 + y_3 + y_4 + \text{etc.}}{\text{number of vertical lines}}. \quad . \quad . \quad . \quad (42)$$

and the mean effective pressure is then determined by multiplying y_m by the scale of the spring.

An indicator diagram similar to that shown in Fig. 76 is occasionally obtained. The small loop on the end represents *negative* work, since the pressure of the steam which

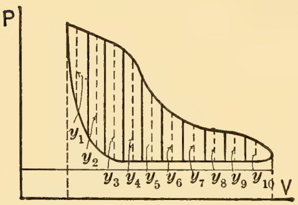


Fig. 75.

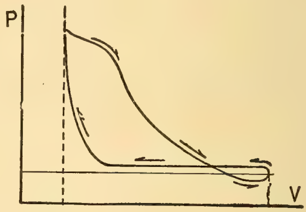


Fig. 76.

does work upon the piston is *lower than* that which resists the return of the piston. When using a planimeter, this area is automatically subtracted from that of the rest of the diagram, but care should be taken to see that this is also done when the method of ordinates is used.

ILLUSTRATIVE PROBLEM

1. Determine the I.h.p. of a double-acting steam engine, having a cylinder 8 ins. diameter, length of stroke, 12 ins., running at 100 R.P.M., the mean effective pressure (M.E.P.) on the piston being 45 lbs. Neglect the area of the piston rod.

$$\text{I.h.p.} = \frac{pLan}{33{,}000} = \frac{(p \times a)\text{ lbs.} \times (Ln)\text{ ft. per min.}}{33{,}000\text{ ft.-lbs. per min.}}$$

$$= \frac{(45 \times 8 \times 8 \times .7854)\text{ lbs.} \times \frac{12}{12} \times 100 \times 2)\text{ ft. per min.}}{33{,}000}$$

$$=\frac{2260 \text{ lbs.} \times 200 \text{ ft. per min.}}{33{,}000}$$

$$=14 \text{ nearly.}$$

2. The I.h.p. of a double-acting engine is 14, the R.P.M. $=100$; M.E.P. $=45$ lbs.; length of stroke $=12$ ins. Find the diameter of the cylinder, neglecting area of piston rod.

First determine the *area* of the piston from the formula

$$\text{I.h.p.}=\frac{pLan}{33{,}000} \quad \text{or} \quad a=\frac{33{,}000 \text{ I.h.p.}}{p \times L \times n};$$

$$a=\frac{33{,}000 \times 14}{45 \times 1 \times 100 \times 2}=51.4 \text{ sq.in.}=\frac{\pi d^2}{4};$$

$$d=\sqrt{\frac{51.4}{.7854}}=\sqrt{65.4}=8 \text{ ins. (approx.).}$$

65. Conventional Diagram and Card Factors. It is often necessary to approximate the mean effective pressure obtained in the cylinder of an engine when no indicator diagrams are available. The most common case is when an engine is being designed to carry a certain load and it is desired to determine the necessary cylinder dimensions and speed. If the *probable* mean effective pressure can be determined, the dimensions and speed can be found from the equation,

$$\text{I.h.p. per cylinder end}=\frac{pLan}{33{,}000},$$

by rewriting it

$$\text{I.p.h.}=\frac{pLn \times 0.7854 d^2}{33{,}000},$$

from which

$$d^2=\frac{33{,}000 \text{ I.h.p.}}{0.7854\, pLn}. \quad \text{(43)}$$

Since n is equal to revolutions per minute for one cylinder end, the product of L by n must be equal to half the piston

speed of the engine, and a proper value of this product can be chosen for substitution in the equation. If a proper value for p can then be predicted the only unknown remaining will be the diameter d, and this can be found by solving the equation.

The prediction of the mean effective pressure is made either by drawing upon recorded experience in the form of values obtained in similar engines previously constructed or by means of what is known as a **conventional indicator diagram.**

The conventional diagram is drawn with upper and lower pressures equal to those expected in the case of the real engine, and all expansions and compressions are drawn as rectangular hyperbolas. The equation of the rectangular hyperbola is

$$P_1V_1 = P_nV_n, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

in which subscript 1 indicates initial conditions and subscript n represents any later conditions with the same material in the cylinder. This law is assumed because it is the simplest and, as a rough average, gives values as close to those actually attained as do any of the more complicated laws.

The diagram may be drawn as nearly as possible like the one which the engine may be expected to give or it may be drawn with various simplifications which remove it more and more from the approximation to an actual indicator diagram. In any case, the mean effective pressure is determined from this diagram and this value is then multiplied by a corrective factor, the value of which has been determined by experience. This corrective factor is called the **diagram factor** or **card factor** and it is really the ratio of the area of the diagram the engine would really give to the area of the conventional diagram used.

The simplest form of conventional diagram is drawn

by neglecting the clearance volume and has the shape shown in Fig. 77. The upper line is drawn horizontal at a height representing the highest pressure expected and of such a length (compared with the length of the diagram) as will approximately represent the fraction of the stroke at which cut-off is to occur in the real engine. The expansion curve is then drawn in as a rectangular hyperbola and extended until the end of the diagram is reached. The next line is drawn vertical and the lower line of the diagram is drawn horizontal at a height representing the pressure expected in the space into which the engine is to exhaust.

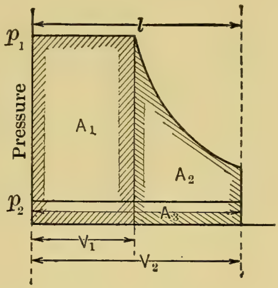


Fig. 77.—Conventional Indicator Diagram.

This simple diagram can be divided into the three areas shown and the value of the work represented by these areas can be determined from the equations given below, the first and last of which should be self-evident from what has preceded. The equation for the work represented by area A_2 can be determined very easily by means of integral calculus. The equations are,

$$A_1 \text{ represents } P_1V_1 \text{ ft.-lbs.};$$

$$A_2 \text{ represents } P_1V_1 \log_e \frac{V_2}{V_1} = P_1V_1 \log_e r \text{ ft.-lbs.},$$

and

$$A_3 \text{ represents } P_2V_2 \text{ ft.-lbs.},$$

in which P represents pressure in pounds per square foot and V represents volume in cubic feet.

The total area is then equal to the sum $A_1+A_2-A_3$ and the net work is equal to a similar sum of the right-

hand members given above. The net work must also equal the mean-effective pressure P_m multiplied by the total volume change, so that

$$P_mV_2=P_1V_1+P_1V_1\log_e r-P_2V_2, \quad . \quad . \quad . \quad (45)$$

and

$$P_m=P_1\frac{V_1}{V_2}+P_1\frac{V_1}{V_2}\log_e r-P_2 \quad . \quad . \quad . \quad . \quad (46)$$

$$=P_1\left(\frac{V_1}{V_2}+\frac{V_1}{V_2}\log_e r\right)-P_2, \quad . \quad . \quad . \quad (47)$$

and substituting $\frac{1}{r}$ for $\frac{V_1}{V_2}$ this gives

$$P_m=P_1\left(\frac{1+\log_e r}{r}\right)-P_2. \quad . \quad . \quad . \quad . \quad . \quad (48)$$

The ratio $\frac{V_2}{V_1}=r$ is called the **ratio of expansion** and its reciprocal, $\frac{V_1}{V_2}=\frac{1}{r}$ is known as the **cut-off ratio.** By the use of this ratio the volume terms can be disposed of and the equation above is obtained. This equation then gives the mean effective pressure in terms of upper and lower pressures and the fraction of the stroke at which cut-off is desired in the real engine and no cylinder dimensions need be known.

Since pressures in steam-engine practice are usually given in pounds per square inch, the equation for mean effective pressure is more useful in the form

$$p_m=p_1\left(\frac{1+\log_e r}{r}\right)-p_2, \quad . \quad . \quad . \quad . \quad . \quad (49)$$

in which p_1 and p_2 and p_m are expressed in pounds per square inch absolute. For convenience in the use of this

equation the values assumed by the bracketed quantity are given for various conditions in Table III.

TABLE III

r	$\frac{1+\log_e r}{r}$	r	$\frac{1+\log_e r}{r}$	r	$\frac{1+\log_e r}{r}$
1.0	1.00	6.0	0.465	16.0	0.236
1.5	0.937	7.0	0.421	17.0	0.226
2.0	0.847	8.0	0.385	18.0	0.216
2.5	0.766	9.0	0.355	19.0	0.208
3.0	0.700	10.0	0.330	20.0	0.200
3.5	0.644	11.0	0.309	21.0	0.192
4.0	0.597	12.0	0.290	22.0	0.186
4.5	0.556	13.0	0.274	23.0	0.180
5.0	0.522	14.0	0.260	24.0	0.174
5.5	0.492	15.0	0.247	25.0	0.169

The values of the mean effective pressures obtained from this form of diagram are very much higher than are to be expected from real engines with the same initial and terminal pressures and the same nominal ratio of expansion. They are therefore corrected by multiplying by the proper diagram factor as selected from Table IV. It is obvious from the range of values given that the selection of a proper value for the factor depends largely on experience, but such experience is quickly gained by contact with real engines and a study of the practical diagrams.

TABLE IV

DIAGRAM FACTORS

Simple slide-valve engine....................	55 to 90%
Simple Corliss engine......................	85 to 90
Compound slide-valve engine...............	55 to 80
Compound Corliss engine...................	75 to 85
Triple-expansion engine....................	55 to 70

66. Ratio of Expansion.—The ratio of expansion used above is sometimes called the *apparent ratio*. It is not the

real ratio of expansion for an engine with clearance. For such an engine the real ratio of expansion is

$$r' = \frac{V_2 + V_{cl}}{V_1 + V_{cl}}, \quad \ldots \ldots \quad (50)$$

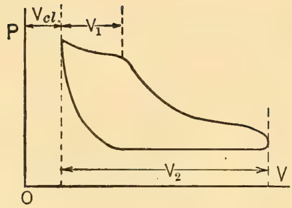


FIG. 78.

in which the symbols represent the volumes indicated in Fig. 78.

The numerical values of r and r' are often very different and care should be used in distinguishing between them. The diagram factors referred to in Table IV are for idealized conventional cards without clearance as shown in Fig. 77.

ILLUSTRATIVE PROBLEMS

1. Given an engine with a stroke of 24 ins. and cut-off occurring at $\frac{1}{3}$ stroke. Steam pressure of 160 lbs. per square inch and back pressure of 16 lbs. Assume diagram factor = 80%. Neglecting clearance, find the probable M.E.P.

$$\text{M.E.P.} = p\left(\frac{1+\log_e r}{r}\right) - p_2 = 160\left(\frac{1+\log_e 3}{3}\right) - 16$$

$$= 160 \times .7 - 16 = 112.0 - 16 = 96 \text{ lbs.}$$

Hence probable M.E.P. = .80 × 96 = 76.8 lbs.

2. A given double-acting engine indicates 75 I.h.p. under the following conditions:

Cut-off at 20%; steam pressure, 140 lbs. per square inch absolute; piston speed, 600 ft. per minute; back pressure, 2 lbs. per square inch absolute.

Assume a diagram factor for this type of engine equal to 85%; and neglecting clearance, find a convenient size of the cylinder (diameter and stroke).

Solution.

$$r=\frac{1}{.20}=5;$$

$$p_m=p\left(\frac{1+\log_e r}{r}\right)-p_2=140\left(\frac{1+\log_e 5}{5}\right)-2=140(.522)-2$$

$$=73.1-2=71.1 \text{ lbs. per sq.in.}$$

Diagram factor $=85\%$. Hence probable

$$\text{M.E.P.}=71.1\times .85=60.4 \text{ lbs.}$$

Therefore, since

$$\text{I.h.p.}=\frac{pLan}{33{,}000}\cdot \quad a=\frac{75\times 33{,}000}{60.4\times 600}=68.3 \text{ sq.in.}$$

$$d=9\tfrac{1}{2} \text{ ins. (approx.);}$$

and since $\quad 2Ln=600$, assume $L=1$ ft.

hence $\quad n=300$ R.P.M.

The engine is rated 9.5×12 ins., running at 300 R.P.M.

67. Determination of Clearance Volume from Diagram. It was shown in a preceding paragraph that the clearance volume of a cylinder must be known in order to draw the line of zero volumes on the indicator diagram. This volume can be determined accurately for any real engine by weighing the quantity of water required to fill the clearance space, but this procedure is often impossible and an alternative, though approximate, method is often resorted to.

This method is graphical and depends upon the assumption of the law of expansion and compression. As in the case of the conventional diagrams, expansion and compression are assumed to follow rectangular hyperbolas.

It is a property of this curve that diagonals such as *aa* and *bb* drawn for rectangles with their corners on the

curve all pass through the origin of coordinates as shown in Fig. 79.

If two points *a* and *c* are selected on the expansion curve of a real diagram and a rectangle is drawn upon them as shown in Fig. 80, the diagonal *bd* extended will pass through the origin of coordinates, if the expansion follows the assumed law. The point at which this diagonal cuts the zero pressure line must therefore be the point through which the vertical line of zero volume is to be drawn.

If the original assumption were correct, this construction would give the same point when different locations of the points *a* and *c* were chosen and when used on the

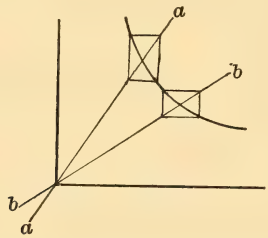


Fig. 79.—Rectangular Hyperbola.

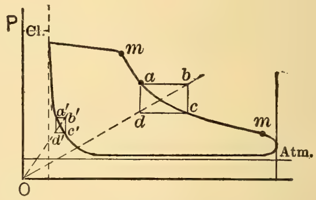


Fig. 80.

compression as well as on the expansion line. In reality it will generally give as many different locations for the origin as are chosen for the rectangle *abcd*. It is customary to construct this rectangle of fair size and to locate it near the center of the expansion curve.

68. Diagram Water Rate. When an engine is run on saturated steam, part of the steam supplied is generally condensed upon the cold metal walls surrounding it. The indicator diagram therefore shows the volumes assumed by the mixture of steam and liquid water in the cylinder, but, since the volume occupied by the liquid is negligible, it may be assumed to show the volumes occupied by the part of the mixture which exists in vaporous form.

Assuming that the vapor is saturated, the volume occupied by one pound at various pressures can be found from the steam tables and, therefore, the weight existing in the cylinder can be calculated. The weight of steam determined in this way is known as the **indicated steam**, the **diagram steam** or the **diagram water rate.**

The diagram water rate is generally determined for a point such as z in Fig. 81 just after cut-off, though some engineers prefer to use a point nearer the lower end of the expansion curve. The volume occupied by the steam contained in the cylinder at point z is equal to V_z and its weight can be determined by dividing this volume by the specific volume $\mathbf{V}_z$ for the existing pressure P_z. Thus, calling the weight of steam in the cylinder w_z,

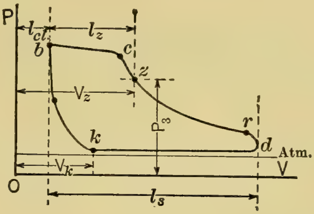


FIG. 81.

$$w_z=\frac{V_z}{\mathbf{V}_z}. \qquad (51)$$

This quantity of steam is a mixture of cylinder feed and clearance or cushion steam and the weight of the latter must therefore be subtracted from w_z to obtain the weight of cylinder feed w_f. Assuming the cushion steam dry and saturated at the point k, the weight of cushion steam is

$$w_k=\frac{V_k}{\mathbf{V}_k}, \qquad (52)$$

so that the weight of cylinder feed per cycle as shown by the diagram at the point z is

$$w_f=w_z-w_k=\frac{V_z}{\mathbf{V}_z}-\frac{V_k}{\mathbf{V}_k}. \qquad (53)$$

The formula is generally modified to give the steam consumption per indicated horse-power hour, instead of

per cycle, and it is also expressed in different terms as a matter of convenience.

For this purpose let

y_{cl} = clearance volume divided by piston displacement per stroke

$= \dfrac{l_{cl}}{l_s}$;

y_z = piston displacement to point z divided by piston displacement per stroke

$= \dfrac{l_z}{l_s}$;

y_k = piston displacement to point k divided by piston displacement per stroke

$= \dfrac{l_k}{l_s}$;

a = area of piston in square inches;

p = mean effective pressure in pounds per square inch;

L = stroke in feet; and

n = number of cycles per minute.

The piston displacement is then $\dfrac{a}{144}L$ cubic feet and the volumes at z and k are given by

$$V_z = \left(y_z \times \frac{aL}{144}\right) + \left(y_{cl} \times \frac{aL}{144}\right),$$

and

$$V_k = \left(y_k \times \frac{aL}{144}\right) + \left(y_{cl} \times \frac{aL}{144}\right).$$

Substituting these values in Eq. (53) gives the indicated cylinder feed per cycle as

$$w_f = \frac{aL}{144}\left(\frac{y_z + y_{cl}}{\mathbf{V}_z} - \frac{y_k + y_{cl}}{\mathbf{V}_k}\right). \quad . \quad . \quad . \quad . \quad . \quad (54)$$

Multiplying by the number of cycles per hour $(60 \times n)$ and dividing by the indicated horse-power, $\dfrac{pLan}{33{,}000}$, gives

the diagram water rate, or steam shown by the diagram per I.h.p. hour as

$$w_d = \frac{13{,}750}{p}\left(\frac{y_z + y_{cl}}{V_z} - \frac{y_k + y_{cl}}{V_k}\right), \quad . \quad . \quad . \quad (55)$$

in which form the equation involves only values which can be determined directly from the diagram without any knowledge of the engine dimensions.

The value obtained for w_d will vary as the location of points z and k are varied because of the quality changes occurring during expansion and compression, and it is obvious that the diagram water rate is in no sense an accurate measure of the *real* water rate of the engine. It is, however, often useful for comparison with the real water rate, the ratio giving an indication of the loss by condensation and leakage.

Average values for real water rates are given in Chapter XI.

ILLUSTRATIVE PROBLEM

Given the diagram shown in Fig. 82 and the following data from an actual test, find the diagram water rate for point c, and for point n. Double-acting steam engine having:

Average piston area = 28.9 sq.-in.;

Length of stroke = 8 in.;

R.P.M. = 237; I.h.p. = 8.75;

M.E.P. = 31.6 lbs.;

Clearance = 13%; Beginning of compression = 29%;

Weight of condensate per hour = 371 lbs.;

Quality at throttle = 95%;

Sp. vol. at c = 7.8;

Sp. vol. at n = 12.57;

Sp. vol. at K = 38.4 cu.ft. per lb.;

Assume x_k = 100%.

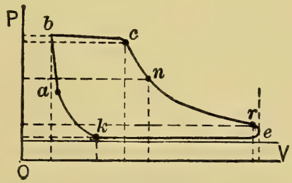


FIG. 82.

Solution. Substitution in Eq. (55) gives

$$(W_d)_c = \frac{13{,}750}{p}\left(\frac{y_c+y_{cl}}{\mathbf{V}_c} - \frac{y_k+y_{cl}}{\mathbf{V}_k}\right)$$

$$= \frac{13{,}750}{31.6}\left(\frac{0.39+0.13}{7.8} - \frac{0.29+0.13}{38.4}\right)$$

$$= 24.2 \text{ lbs. per I.h.p. per hour at point } c.$$

$$(W_d)_n = \frac{13{,}750}{31.6}\left(\frac{y_n+y_{cl}}{V_n} - \frac{y_k+y_{cl}}{V_k}\right)$$

$$= \frac{13{,}750}{31.6}\left(\frac{0.638+0.13}{12.57} - \frac{0.29+0.13}{38.4}\right)$$

$$= 21.83. \text{ lbs. per I.h.p. per hour at point } n.$$

$$\text{Real water rate} = \frac{371}{8.75} \times 0.95 = 40.2 \text{ lbs.}$$

69. $T\phi$-diagram for a Real Engine. In Chapter VI the $T\phi$-diagrams of the various ideal cycles were given and attention was called to the fact that these diagrams were particularly useful, because they showed certain things which were not apparent from the more common *PV*-diagrams.

It has been customary for many years to draw $T\phi$-diagrams for real engines by "transferring" the *PV*-diagram to $T\phi$-coordinates, and various analytical and graphical methods have been developed for this purpose. There are certain unavoidable errors in all the methods used for drawing these diagrams, and the expansion curve is the only one of all the lines finally obtained which has any claim to accuracy. Even this curve is generally incorrectly interpreted, because a knowledge of the exact weight of clearance steam is necessary for an accurate interpretation and such knowledge is never available.

Under the circumstances it seems unnecessary to consider in this book the rather complicated details involved in the construction of $T\phi$-diagrams purporting to show the behavior of steam in real engines.

70. Mechanical and Thermal Efficiencies. The method of obtaining the indicated horse-power from the indicator diagram has been given in preceding paragraphs. In the real engine this power is not all made available at the shaft, because some of it is used in driving the engine against its own frictional resistance. Calling the power lost in this way the **friction horse-power**, it follows that

$$\text{I.h.p.} = \text{F.h.p.} + \text{D.h.p}, \quad . \quad . \quad . \quad . \quad . \quad (56)$$

in which

I.h.p. = indicated horse-power determined from the real indicator diagram;

F.h.p. = friction horse-power, i.e., power required to drive engine; and

D.h.p. = developed horse-power, i.e., power made available at shaft. This is sometimes called brake horse-power and abbreviated B.h.p.

The **developed horse-power** is therefore always less than the indicated horse-power. The better the construction of the engine the smaller the friction loss, and the measure of this loss is usually given in the form of an efficiency. It is called the **mechanical efficiency,** and is defined by the equation

$$\text{Mech. eff.} = \frac{\text{D.h.p.}}{\text{I.h.p.}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

The developed or brake horse-power can generally be determined experimentally in one way or another so that it becomes possible to evaluate numerator and denominator of Equation (57) and to determine the mechanical efficiency of the engine. Methods of determining the developed horse-power are considered in Chapter XX. Values of mechanical efficiency range from about 80 per cent in the case of poorly designed and poorly adjusted horizontal engines to about 95 per cent in the case of the best vertical designs.

The efficiency determined by dividing energy made available by heat supplied is known as the **thermal efficiency.** There are two possible thermal efficiencies, one based on the indicated power and the other on the developed power. The former is called the *thermal efficiency on the indicated horse-power* or the *indicated thermal efficiency;* the other is known as the *thermal efficiency on the developed horse-power* or the *developed thermal efficiency.* Obviously

$$\text{Dev. ther. eff.} = \text{Mech. eff} \times \text{Indic. ther. eff.} \quad . \quad (58)$$

The heat supplied may be assumed in two different ways; it may be taken as the total heat above 32° F. in the steam supplied the engine, or it may be taken as this value less the heat of the liquid corresponding to exhaust temperature. The second method is preferable, since it is reasonable to assume that the exhaust steam can be condensed to water at the same temperature or raise an equal weight of water to that temperature and that this water can be pumped to the boiler with the heat of the liquid corresponding to this temperature. This is practically parallel to the assumption made in treating the theoretical cycles.

The thermal efficiencies are then

$$\text{Indic. ther. eff.} = \frac{\text{I.h.p.} \times 2545}{\text{Heat above liquid at exhaust temp. per hour}}, \quad . \quad (59)$$

and

$$\text{Dev. ther. eff.} = \frac{\text{D.h.p.} \times 2545}{\text{Heat above liquid at exhaust temp. per hour}}. \quad . \quad (60)$$

Values of the indicated thermal efficiency range from about 5 per cent in ordinary practice with small engines to about 25 per cent in the best large engines. Values as low as 1 per cent are not uncommon with small, poorly designed and poorly operated engines.

The actual performance of the cylinder of an engine is sometimes compared with the ideal possibilities as indicated by the Clausius and the Rankine cycles. The ratio of the work obtained in the real engine to that which could be obtained from the same quantity of heat with a Rankine or Clausius cycle is a measure of the performance of the real cylinder. This ratio is variously designated as *cylinder efficiency*, *indicated efficiency*, *relative efficiency*, etc. Its values range from less than 40 per cent to over 80 per cent, the highest recorded value being just over 88 per cent.

PROBLEMS

1. Using Table I, Chapter I, plot the specific heat of water between the range of temperatures of 20° F. and 300° F. for the intermediate values given. By the ordinate method for finding the mean height of an indicator diagram, determine the mean or average specific heat over this range.

2. A double-acting engine is required to give 50 I.h.p. under the following conditions:

Cut-off = 25%;
Steam pressure = 150 lbs. per square inch absolute;
Back pressure = 16 lbs. per square inch absolute;
Piston speed = 540 ft. per minute.

If the diagram factor for this type of engine is 75%, find the diameter of the cylinder and select the stroke and R.P.M.

3. Assume a single-acting engine with 10-in. diameter and 12-in. stroke, 10×12 ins., to have cut-off occur at various points between 10% and 50% of stroke. Assume also the pressures, speed, and card factor as given in Prob. 2. Find the probable I.h.p. at different cut-offs.

4. Given an 18×24-in. engine running at 120 R.P.M.

Back pressure = 2 lbs. per square inch absolute;
Clearance = 10%;
Cut-off = 40%;
Diagram factor = 85%.

Supposing cut-off to remain constant, find the I.h.p.'s corresponding to steam pressure of 50, 90, and 130 lbs. per square inch absolute.

5. Find the weight of dry steam which must be supplied per I.h.p. hour for each case of the previous problem, assuming the

quality at cut-off to be 80%. Assume compression pressure to be 30 lbs. absolute and that steam is dry and saturated at end of compression.

6. Find the quality of steam at cut-off in a cylinder, in which the piston displacement is 0.1278 cu.ft.; clearance = 10%; cut-off at 25% stroke; steam pressure at cut-off, 115 lbs. per square inch absolute. and weight of steam in the cylinder at cut-off = 0.012 lb.

Note. $\text{Quality} = \dfrac{\text{Actual vol.}}{\text{Weight} \times \text{Sp. vol.}}$ for the given pressure.

7. The piston displacement of a certain engine is 0.2 cu.ft. What weight of steam is in the cylinder at release where quality is 90%, and pressure is 25 lbs. per square inch absolute, if the clearance is 10%, and release occurs at 95% of the stroke?

8. Find the weight of cushion steam in a 6×6 in. engine in which clearance = 15%; compression begins at 85% of the return stroke; back pressure is 14.7 lbs. per square inch absolute, and the quality of the cushion steam at the beginning of compression is 95%.

9. Find the pressure and quality at the end of the compression line of the previous problem, assuming it to be adiabatic.

10. An 8×10 in. engine running at 300 R.P.M. is double-acting, and cuts off at 15% of the stroke at a pressure of 120 lbs. per square inch absolute. It has a steam consumption of 35 lbs. per I.h.p.-hour. The compression begins at 60% of the return stroke with a quality of unity and a back pressure of 5 lbs. per square inch absolute. Clearance = 10%.

If this engine delivers 27 H.P., and has a mechanical efficiency of 90%, what is the quality at the point of cut-off?

11. In the previous problem, assume release to occur at 90% of the stroke with an absolute pressure of 30 lbs. per square inch. What is the quality at this point?

12. A certain engine gives one horse-power hour at the shaft for every 20 lbs. of steam supplied. The steam has an initial pressure of 150 lbs. absolute and is dry and saturated when it arrives at the engine. The back pressure against which steam is exhausted is 4 lbs. absolute.

(*a*) Find the thermal efficiency of this engine on the developed or shaft horse-power.

(*b*) If the mechanical efficiency of the engine is 90%, what is the value of the thermal efficiency on the indicated horse-power?

CHAPTER IX

COMPOUNDING

71. Gain by Expansion. The cycle which gives a rectangular *PV*-diagram is the least economical of all the ideal cycles described in Chapter IV. This comes from the fact that none of the heat stored in the steam is converted into work when this cycle is used. Thus, if the cylinder shown by full lines in Fig. 83 operate on this cycle and be of such size that it will receive just one pound of steam per cycle, it makes available an amount of work represented by the area *abcd*. The positive work done by the steam upon the piston is the equivalent of the external latent heat of vaporization while no use is made of the heat stored in the steam. This stored heat is removed *as heat* during the condensation and exhaust, which give the lines *bc* and *cd*.

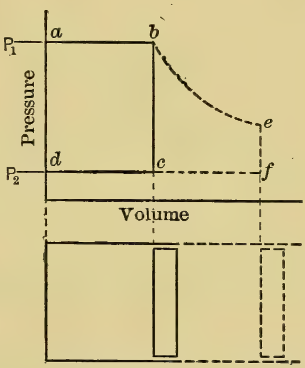


Fig. 83.

If a piece be added to the cylinder as indicated by the dotted lines, the same quantity of steam will make more heat available by expanding after cut-off, as shown by the curve *be*, the net work in this case being represented by the area *abefd* instead of by the smaller area *abcd*. But the heat supplied is the same in both cases, namely that required to form one pound of steam at the pressure P_1, so that the use of a large cylinder and the incomplete ex-

pansion cycle results in the development of more work than can be obtained with the rectangular cycle from the same amount of heat.

Obviously it would be theoretically advantageous to add still more to the length of the cylinder and allow the expansion to continue to back pressure, giving the complete expansion cycle as shown in Fig. 84, thus obtaining the maximum quantity of work at the expense of the heat stored in the steam supplied the cylinder. Practically, it is found inadvisable to continue the expansion to such a de-

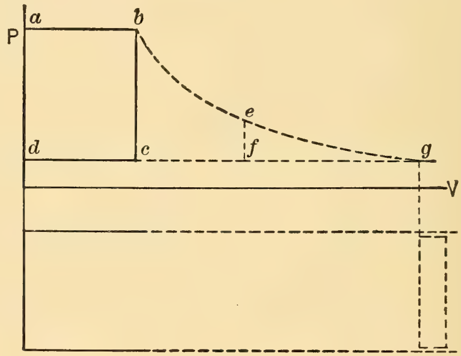


FIG. 84.

gree in reciprocating steam engines, because at low pressures the volume increases very rapidly for small pressure drops. Thus a great increase is necessary in the size of the cylinder if the last part of the expansion is to be completed, but the amount of work obtained is comparatively small, as shown by the small height of the long toe thus added to the diagram. This may result in an actual loss, because the increased friction losses of the very large cylinder may more than balance the small increase of net work gained by its use. It thus results that, in every real reciprocating engine, there is some point beyond which it is not economical to carry the expansion, and the incomplete expansion

cycle is therefore approximated in such engines rather than the cycle with complete expansion.

Viewing the matter from another angle, a cylinder of a certain size may be assumed as shown in Fig. 85. The use of the rectangular cycle *abcd* in this cylinder will make available the maximum quantity of work possible with the upper and lower pressures chosen. If cut-off be made to occur earlier as at b', the expansion $b'c'$ will result in a loss of the quantity of work obtained, as shown by the area $b'bc'$, but the steam used per horse-power will be less, so that there will be a gain in steam economy. Putting the cut-off still earlier will cause a still greater loss of work obtained from a cylinder of the chosen size, but theoretically will result in greater economy of steam.

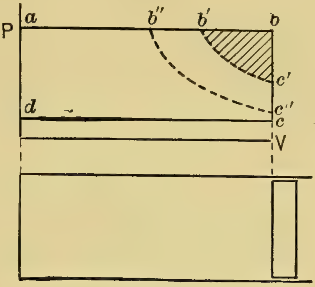


FIG. 85.

Summing up, it may be said that the greater the ratio of expansion the greater should be the economy in the use of steam on a theoretical basis.

The lower pressure is set in real engines by the pressure in the space into which the engine is to exhaust. If the engine is to be operated non-condensing, the atmospheric pressure determines the lowest possible exhaust pressure; if the engine is to be operated condensing, the exhaust pressure is set by the lowest pressure which can be economically maintained in the condenser.

There is thus a real limit to the extent to which expansion can be carried in any real engine with a given initial pressure. A certain drop must exist at the end of the diagram, for reasons already explained, and an expansion line drawn backward from the top of the line representing this drop will give the earliest possible cut-off

which can be used in the engine with a given initial pressure.

The ratio of expansion can be further increased, however, by raising the initial pressure as indicated by the dotted lines in Fig. 86, and the limit in this direction would come with the inability of materials of construction to withstand the resulting strains.

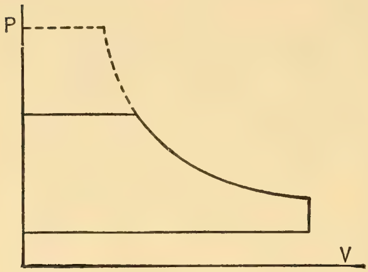


FIG. 86.

These conclusions drawn from the facts developed above must all be modified in the case of real engines, because of the effect of cylinder condensation. This has been shown to increase as the cut-off is made earlier and as the pressure (and therefore the temperature) range in a cylinder is increased. There is, therefore, a limit beyond which it is not advisable to carry the ratio of expansion in a single cylinder.

Experience has shown that in the case of engines expanding steam entirely in one cylinder (*simple engines*) *the best commercial results are obtained when* (a) *they are operated noncondensing,* (b) *the initial pressure is between* 80 *and* 100 *lbs. per square inch for the simpler forms of valves and below* 125 *lbs. for the better forms, and* (c) *the point of cut-off is at about* $\frac{1}{4}$ *stroke with the simpler valves and at* $\frac{1}{5}$ *to* $\frac{1}{4}$ *stroke with the better forms of valves. Strictly, these statements apply only to engines of the types so far treated. The Una-flow engine, described later, has entirely different characteristics.*

72. Compounding. If the ratio of expansion of ordinary types is to be increased above the values just given, some means must be used for the reduction of loss by condensation. This loss can be reduced by decreasing the surface exposed to high-temperature steam and by de-

creasing the temperature range in a cylinder. Both of these results can be achieved by what is known as **compounding.**

Assume that it is deemed advisable to produce a cycle similar to that shown in Fig. 87 (clearance neglected) and that in order to obtain *high steam economy* (*low water rate*) the ratio of expansion chosen is very much greater than four. No gain in economy would result from such excessive expansion in a single cylinder, in fact there would be a well-defined, unavoidable loss. But suppose that the high-pressure steam is admitted to a small cylinder such as that shown and is expanded to the point f, is then exhausted as shown by fg into the larger cylinder along gf and then expanded to the point c in the larger cylinder. The cycle produced is the same as that which would have been obtained by expanding entirely in one cylinder, but the surface of the clearance space of the high-pressure (H.P.) cylinder, which is exposed to high-pressure steam is smaller than it would be in a cylinder of the size required to hold the steam when fully expanded and, moreover, the lowest temperature to which it is subjected is that corresponding to the pressure at f instead of the much lower temperature corresponding to the pressure at d.

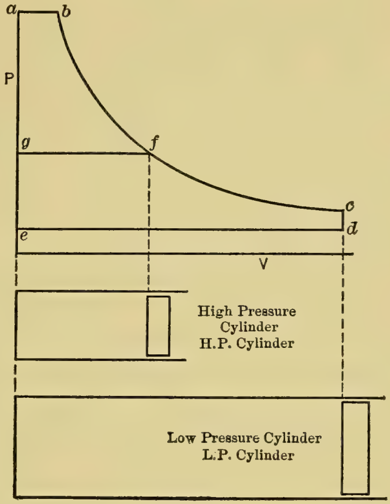


Fig. 87.

The condensation which would occur in the H.P. cylinder would obviously be less than that which would result from

the use of one large cylinder and, remembering that the greater part of the heat given up during condensation is received again by the steam during exhaust, it is obvious that approximately this same quantity of heat can again be given to the low-pressure cylinder walls. Thus, although there are two cylinders in which condensation and re-evaporation occur, and although the sum of the heat given to the walls of the high-pressure cylinder and that given to the walls of the low-pressure cylinder might be greater than that given to the walls of a single cylinder under similar conditions, the use of two cylinders results in a considerable saving because loss in the high-pressure cylinder is practically wiped out by the exhaust of the heat concerned into the low-pressure cylinder.

If the loss by radiation and conduction from the high-pressure cylinder be neglected, the result of the use of two cylinders is practically to limit the loss by condensation and re-evaporation to that occurring in the low-pressure cylinder. As the ratio of expansion in this cylinder is in the neighborhood of that common in simple engines, or even less, and as the temperature range is small, the net loss is also small.

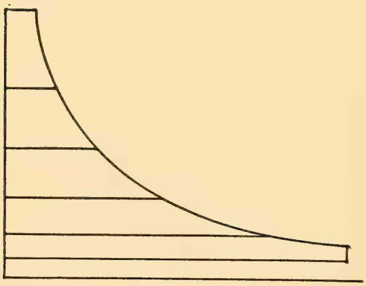


Fig. 88.

It is obvious that the smaller the surface of the high-pressure cylinder can be made, and the smaller the temperature range in a single cylinder, the smaller will be the net loss by cylinder condensation and re-evaporation. A saving should therefore be effected by using more than two cylinders, and it is not inconceivable that five or more might be used. The result of using five cylinders is shown in Fig. 88, and it is evident that the clearance surfaces exposed to

high temperatures, the temperature ranges per cylinder and the ratios of expansion per cylinder are all small. The gain in economy should therefore be correspondingly great.

There are *two limits to the possible multiplication of cylinders* in this way.

(1) As the number increases the radiating surface and therefore the heat lost by radiation increases. The extent of this effect can be appreciated by noting that every cylinder with the exception of the low-pressure cylinder is really an unnecessary addition, because the cycle could be produced entirely in the low-pressure cylinder. On the other hand, the surfaces of cylinders which operate at high temperature are small as compared with that which would be exposed to this temperature if the entire cycle were produced in the low-pressure cylinder.

(2) As the number of cylinders is increased, the first cost, the complexity and the cost of lubrication and attendance are all increased so that, for each installation, some number will be found beyond which the interest on the investment and the added cost of operation and maintenance would more than balance the saving of fuel.

The second limit mentioned is the more important commercially, as it is the first one reached. For ordinary operating conditions in stationary power plants expansion in two cylinders generally gives the most economical results. The total ratio of expansion is generally between 7 and 16, that is, the volume of steam at release in the L.P. cylinder is from 7 to 16 times the volume at cut-off in the H.P. cylinder. For large pumping stations and large marine installations, expansion in three cylinders is generally considered the most economical, and total ratios of expansion of 20 or more are used. Four and five cylinders have been used, but the resultant gains do not seem to warrant any extensive installation of such units.

Engines using more than one cylinder for the expansion

of steam in the way just described are called **multi-expansion engines**, or **compound engines**, and the use of multi-expansion is spoken of as compounding. Custom has almost confined the use of the term compound engine to those in which only two cylinders are used in series as indicated in Fig. 89, and such engines are often spoken of as $2x$ engines.

Engines in which three cylinders are used in series are called *triple-expansion* or $3x$ engines. With four and five cylinders in series the engines are known as *quadruple* or $4x$ and *quintuple* or $5x$, respectively.

In the case of triple-expansion engines of large size,

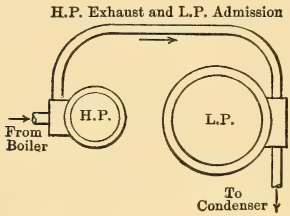


Fig. 89.

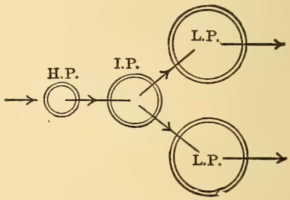


Fig. 90.

the volume of the low-pressure cylinder required generally becomes so great that it is found economical to use two low-pressure cylinders instead of one. The flow of steam in such an engine is represented diagrammatically in Fig. 90. This type is known as a *four-cylinder, triple-expansion engine*.

All multi-expansion engines are generally operated condensing, and the choice of type is determined partly by the character of work to be done and partly by economical considerations. In all cases the boiler pressure must be chosen to suit the type of engine used. The pressures ordinarily used with the different types are given in Table V.

TABLE V

BOILER PRESSURE COMMONLY USED

Type of Engine.	Boiler Pressure. Pounds per Sq.in. Gauge.
Simple	80 to 125
High-speed compound	100 to 170
Low-speed compound	125 to 200
Triple expansion and higher	125 to 225

73. The Compound Engine. The term compound engine will be used hereafter in the commercial way as referring to a $2x$ engine. Such engines may roughly be divided roughly into two types, *receiver* and *non-receiver* engines. The latter are often called Woolf engines, after the man who first used this construction.

A receiver engine has a vessel known as a receiver located between the two cylinders and so connected with them that the high-pressure cylinder exhausts into the receiver and the low-pressure cylinder draws its steam from the receiver. By using a receiver the cylinders are made independent of each other so far as steam events are concerned; the high-pressure cylinder can exhaust at any time without reference to the events occurring in the low-pressure cylinder.

A Woolf type has practically no receiver, the high-pressure cylinder exhausting directly into the low-pressure cylinder through the shortest convenient connecting passage. As the high-pressure cylinder must exhaust directly into the low-pressure cylinder it follows that cut-off must not occur in the latter until compression starts in the former; i.e., very near the end of the stroke.

An engine with a receiver of infinite size would give a horizontal exhaust line for the high-pressure cylinder and a horizontal admission line for the low-pressure cylinder, since the small amount of steam given to or taken from the

receiver would have no appreciable effect upon the pressure within that vessel. Neglecting throttling losses, the high-pressure and low-pressure cards would therefore fit together as originally indicated in Fig. 86.

With receivers of finite size there are pressure changes during exhaust by the high- and admission to the low-pressure cylinders, and real valves and connections also cause certain throttling losses, so that the lines representing these events are not horizontal nor do they exactly coincide.

A diagrammatic arrangement of the Woolf engine is given in Fig. 91 with idealized diagrams obtained by

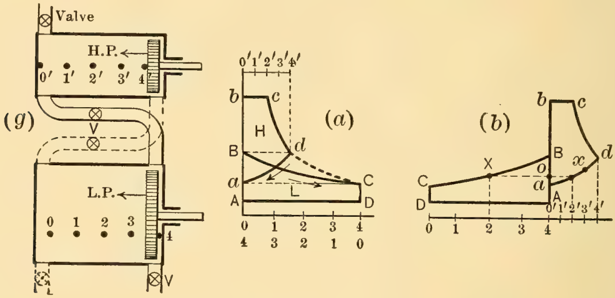


FIG. 91.

assuming hyperbolic expansions, no clearances, and no throttling losses. The pistons must make their strokes together in such engines, but they may move in the same direction, as shown in the figure, or in opposite directions.

The ideal diagram would be that shown at (*a*) by the lines *AbcdCDA*. The idealized high-pressure diagram is *abcda* and the idealized low-pressure diagram is *ABCDA*. The exhaust line *da* of the high-pressure diagram and the admission line *BC* of the low-pressure diagram are produced at the same time. Corresponding points on these two lines represent the common pressures assumed by the steam not yet exhausted from the high-pressure cylinder, the steam in the small connecting passage and the steam

already admitted to the low-pressure cylinder. As the movement of the low-pressure piston opens up volume faster than the high-pressure piston closes up volume, the volume occupied by the steam continues to increase as the low-pressure piston moves out, and its pressure therefore decreases.

The two diagrams are shown back to back at (*b*) in the figure and the horizontal line xX connects corresponding

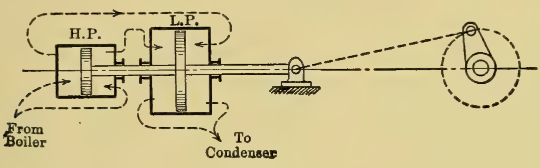


FIG. 92.

points on the exhaust of the high pressure and the admission of the low pressure cylinders respectively.

Compound engines are also divided into two types on the basis of cylinder arrangement. When the axes of both cylinders coincide as shown in Fig. 92 they are called **tandem compounds**. When the axes are parallel as shown in Fig. 89, the engines are spoken of as **cross-compound engines**.

74. Cylinder Ratios. The idealized diagrams of a compound engine with infinite receiver volume are shown in Fig. 93 by *abcd* and *ABCDE*. The height of the high-pressure exhaust line is the same as that of the low-pressure admission line and represents the receiver pressure p_R. The value of the receiver pressure is determined

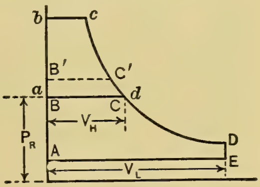


FIG. 93.

by the point chosen for cut-off in the low-pressure cylinder. Thus if cut-off in the low-pressure cylinder is made to occur earlier, as at some point C', the admission line for this

cylinder must move up to $B'C'$ and the receiver pressure must rise correspondingly. The exhaust pressure in the high-pressure cylinder would also rise an equal amount.

Changing the point of cut-off in the low-pressure cylinder also produces another result. As the receiver pressure rises the work area of the high-pressure diagram is obviously decreased, while that of the low-pressure diagram is increased. In a simple engine the area of the diagram becomes smaller the earlier the cut-off, and it should be noted that just the reverse of this occurs in the low-pressure cylinder of a compound engine.

It is evident that the choice of the receiver pressure or of the point of cut-off in the low-pressure cylinder determines the relative areas of the high-pressure and low-pressure diagrams and it also determines the relative size of the two cylinders. The diagram of Fig. 93 shows that late cut-off in the low-pressure cylinder calls for a larger high-pressure cylinder than does early cut-off.

The ratio of the piston displacement of the low-pressure cylinder to that of the high-pressure cylinder is called the cylinder ratio. Designating this ratio by R, and using other symbols as in Fig. 93,

$$R=\frac{V_L}{V_H}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

The cylinder ratios chosen for real compound engines vary greatly in different designs and no given ratio has been proved the best for a given set of conditions. Normal practice gives the average values listed in Table VI, but cylinder ratios as high as 7 have been used with excellent results.

TABLE VI

CYLINDER RATIOS FOR COMPOUND ENGINES

Cylinder ratio	$2\frac{1}{2}$	$3\frac{1}{4}$	4	$4\frac{1}{2}$
Initial pressure (gauge) non-condensing	100	120		
Initial pressure (gauge) condensing	.....	100	120	150

The cylinder ratio to be used in a given case may be determined by any one of several considerations or by a combination of them, the latter being more often the case. Thus it may be deemed desirable to obtain the same amount of work from both cylinders; or to obtain equal temperature ranges; or to have cut-offs occur at the same fraction of the strokes; or to have the same total load on the two piston rods during admission; or to obtain the maximum possible uniformity of turning effort at the crank. The consideration of equal work is generally regarded as the most important.

75. Indicator Diagrams and Mean Pressures. The idealized diagrams for a compound engine with clearance, with incomplete expansion in both cylinders, and without compression are given in Fig. 94. The nominal total ratio of expansion would be $L_L \div l_H$, but the total ratio of expansion taking account of clearance is

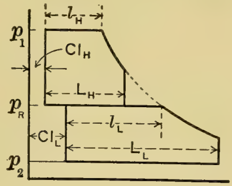


FIG. 94.

$$\text{Total ratio of expansion} = \frac{L_L + Cl_L}{l_H + Cl_H}, \quad . \quad . \quad . \quad (62)$$

and the cylinder ratio is

$$R = \frac{L_L}{L_H}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (63)$$

The mean effective pressures can be found from each of the diagrams in the ordinary way and the indicated horse-power of each cylinder determined therefrom. *The indicated horse-power of the engine is then equal to the sum of the values obtained for the separate cylinders.*

It is often convenient to refer the mean effective pressure of all cylinders to the low-pressure cylinder as though this were the only cylinder acting. In the simple form of diagram, such as that shown in Fig. 93, it is obvious that this could be obtained by measuring the area $AbcDEA$,

dividing by the length AE and multiplying by the scale of the spring, just as though the diagram were all produced in one cylinder with the piston displacement equal to V_L. In the case of the diagrams given in Fig. 94 a similar method could be adopted, or the mean effective pressure of each cylinder could be determined separately and then the equivalent pressure which would give the same result on the low-pressure piston could be determined analytically.

Assume for this purpose that the mean effective pressure of the high-pressure is equal to p_H pounds per square inch, that the mean effective pressure of the low-pressure cylinder is equal to p_L and that the cylinder ratio is R. The strokes of all cylinders of a multi-expansion engine are generally equal, so that the piston areas are in the same ratio as the cylinder volumes (piston displacements). In the case of a $2x$ engine, therefore, the area of the low-pressure piston is R times as great as that of the high-pressure piston, and the pressure required on the low-pressure piston to do the same work as that done by pressure p_H on the high-pressure piston will be $\frac{p_H}{R}$.

In the case of a $2x$ engine therefore the total M.E.P. referred to the low-pressure cylinder is

$$p_R = \frac{p_H}{R} + p_L. \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

This mean effective pressure acting on the low-pressure piston only would give the same indicated horse-power as is obtained with the two cylinders of the engine.

In designing compound engines it is customary to determine the size of the low-pressure cylinder as though it were to do all the work expected of the engine by receiving steam at the highest pressure available and exhausting it at the lowest. The mean effective pressure which would thus be assumed to exist is the referred value p_R just explained. Having found the size of the low-pressure cylinder

and the value of the referred M.E.P. the size of the high-pressure cylinder can be determined so that the work done by each cylinder will be just half of the total for which the engine is being designed. This size will have to be such that the high-pressure mean effective pressure referred to the low-pressure cylinder (i.e., $p_H \div R$) is equal to half the total mean effective pressure referred to that cylinder. That is, the size will have to be so chosen that

$$\frac{p_H}{R}=\frac{p_R}{2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

ILLUSTRATIVE PROBLEM

A double-acting compound engine is capable of developing 500 I.h.p. The stroke is 18 ins.; revolutions per minute, 175; mean effective pressure referred to L.P. piston, 45 lbs. per square inch; cylinder ratio, $3\frac{1}{2}$. Find cylinder diameters.

From

$$\text{I.h.p.}=\frac{pLan}{33{,}000},$$

$$a_{\text{L.P.}}=\frac{500\times 33{,}000}{45\times 1.5\times 175\times 2}=700 \text{ (approx.)};$$

so that

$$d_{\text{L.P.}}=\sqrt{\frac{700}{.785}}=30 \text{ ins. (approx.)},$$

with the cylinder ratio equal to $3\frac{1}{2}$,

$$a_{\text{H.P.}}=\frac{700}{3.5}=200 \text{ sq.ins.},$$

$$d_{\text{H.P.}}=\sqrt{\frac{200}{.785}}=16 \text{ ins. (approx.)}.$$

76. Combined Indicator Diagrams. When a compound engine is indicated, the diagrams of the two cylinders as drawn by the indicator are not directly comparable. The scales of pressure and volume are different on the two diagrams, and correction must be made for this fact before the

diagrams can be compared. It is customary to do this and to draw the average high-pressure and low-pressure diagrams on the same set of coordinates in order to determine how well they approximate the ideal diagram that would be obtained in one cylinder operating between the extreme limits of pressure.

Diagrams approximating those that would be obtained from high- and low-pressure cylinders are shown at (*h*) and (*l*) respectively, in Fig. 95, and the result of drawing

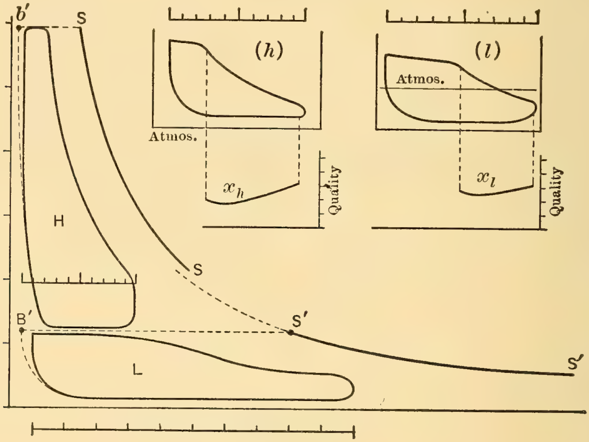


Fig. 95.

both to the same scales is shown at the left of this figure. The curves x_h and x_l show the variations of quality along the two expansion curves.

Drawing the two diagrams to the same scales in this way is known as combining the diagrams and the result is known as a **combined diagram.**

The curves SS and $S'S'$ added to the combined diagram are **saturation curves.** They do not, in general, form a continuous curve, because of the different quantities of steam contained in the two clearances and because any

moisture in the high-pressure exhaust is generally removed in the receiver. The volumes occupied by clearance steam at initial pressures are indicated by the points b' and B' respectively. The lengths $b'S$ and $B'S'$ approximately represent the volumes that would be occupied by cylinder feed when in each cylinder if dry and saturated.

A combined diagram for a triple-expansion engine is shown in Fig. 96. The heavy lines give diagrams constructed so as to represent as nearly as possible what may be expected to occur in the cylinders of such an engine, assuming perfect valve action and hyperbolic expansions and compressions. The dotted diagrams indicate the shapes that would be drawn by indicators applied to the real cylinders. The numerous sharp angles are due to overlapping of events, one cylinder suddenly starting to draw from a receiver while another is exhausting. It will be observed that the dotted diagrams do not contain any of these sharp angles, but that their general outline forms a fair average of them.

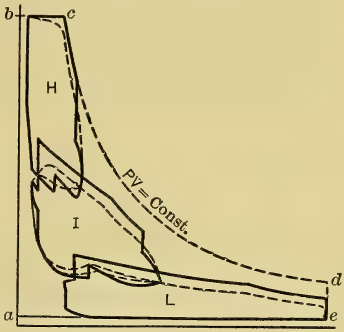


Fig. 96.

The curve cd is a rectangular hyperbola drawn as a continuation of the assumed hyperbolic expansion line of the high-pressure cylinder. The failure of the expansion lines of the other cylinders to fall upon this curve is explained by quality changes, different quantities of clearance steam in the different cylinders and withdrawal of moisture from steam exhausted to receiver before admission to the following cylinder.

PROBLEMS

1. Find the size of the cylinders of a double-acting compound engine, which is to give 600 I.h.p., when using steam at a pressure of 150 lbs. per square inch absolute, and having a back pressure of 2 lbs. per square inch absolute. The cylinder ratio is to be 4, and the total ratio of expansion 12, piston speed 750 ft. per minute, and R.P.M. = 150; diagram factor is 80%.

2. Given a 200 H.P. compound Corliss engine with cut-off in the H.P. cylinder at 60% stroke. Ratio of expansion is 7; clearance is 7%; card factor is 70%; pressure at the H.P. cylinder is 165 lbs. absolute. Find

(*a*) Cylinder ratio;
(*b*) Theoretical and actual M.E.P.;
(*c*) Determine size of four engines, and select the best one.

Note. $$r' = \frac{1 + \%\ \mathrm{Cl}}{\%\ (\mathrm{C.O.}) + \%\ \mathrm{Cl}} \times \text{cyl. ratio.}$$

3. Given a compound engine 18×40 ins., having a stroke of 28 ins. Steam pressure is 165 lbs. per square inch absolute; cut-off in H.P. cylinder occurs at 62% stroke; clearance equals 16%; back pressure equals 5 lbs.; R.P.M. equal 150. Find

(*a*) Cylinder ratio;
(*b*) Ratio of expansion;
(*c*) Actual M.E.P.;
(*d*) I.h.p.

CHAPTER X

THE D-SLIDE VALVE

77. Description and Method of Operation. The simple D-slide valve, shown in place in Fig. 97, is so named because of the similarity of its section to the letter D. It is located in the steam chest, rides back and forth upon its seat and

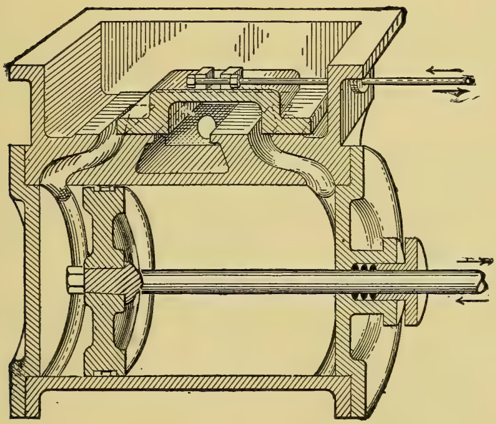


Fig. 97.

serves to connect the two ports alternately with steam and exhaust spaces respectively in order to give the necessary distribution of steam.

The valve has to perform the following functions for each end of the cylinder during each revolution of the engine:

(1) It connects the proper port to the steam space or

steam chest at such a time that steam can enter the cylinder as the piston moves away from the head.

(2) It shuts off this port and thus cuts off the supply of steam when the piston has completed a certain definite fraction of the stroke.

(3) It connects the port with the exhaust cavity shortly before the piston reaches the end of the stroke, thus effecting "exhaust" or "release"; and

(4) It shuts off the port again when the piston has completed the proper fraction of the next stroke, thus trapping in the cylinder the steam which is compressed during the remainder of the stroke.

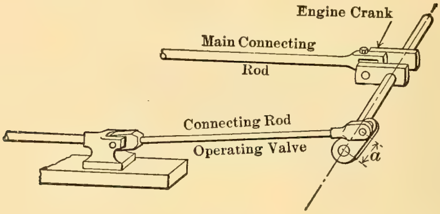


Fig. 98.

It is obvious that the valve must be reciprocated upon its seat and that its motion must be connected with that of the piston in some way so that the proper phase relation may be retained. This could be effected by the system shown diagrammatically in Fig. 98, a small crank operating on the end of a connecting rod giving the valve its short stroke just as the main crank fixes the longer stroke of the piston. Such an arrangement would, however, be very inconvenient with many real engines, as the valve would be located too far from the center line of the cylinder.

It is customary to use what is known as an **eccentric** for the purpose of operating the slide valve. The parts and arrangement of an eccentric, together with an illustration of the way in which it is mounted on the shaft of

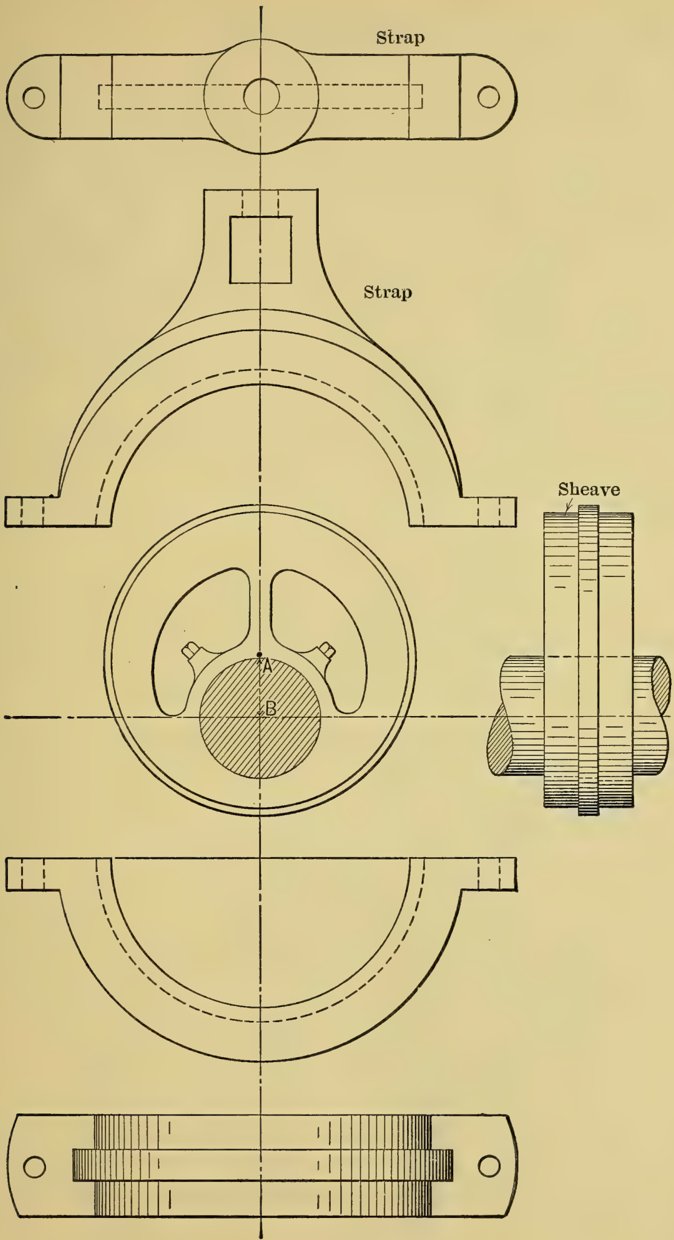


FIG. 99.—Parts of Eccentric.

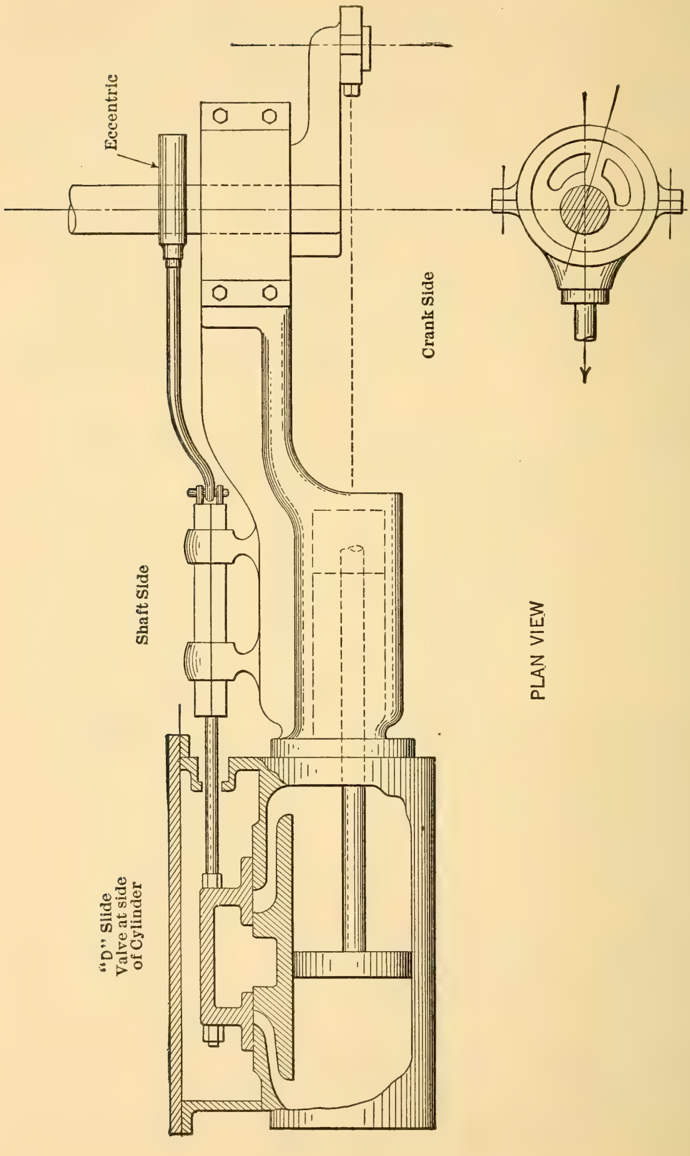


FIG. 100.—Eccentric on Horizontal Engine.

an engine are shown in Figs. 99, 100 and 101. The motion it gives the valve is exactly the same as that imparted by the crank first assumed, and it can easily be shown that it is the exact equivalent of such a crank.

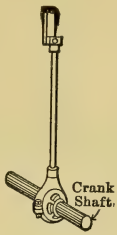


FIG. 101.—Eccentric on Vertical Engine.

Assume, for example, a crank such as that shown in Fig. 98 with a length of arm or throw equal to *a*. If the crank pin is made larger while other parts of the crank remain the same, as shown in Fig. 102, the crank mechanism is not essentially altered; the motion which it would impart to a connecting rod is not changed. If this process of enlarging the pin be continued

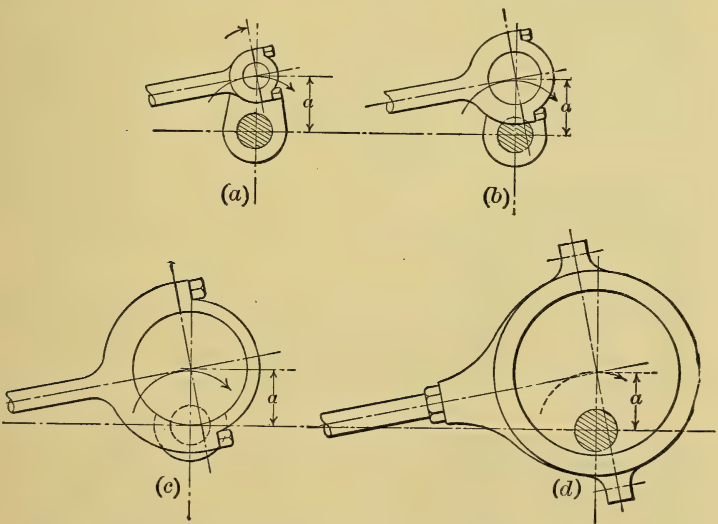


FIG. 102.—Equivalence of Crank and Eccentric.

until the pin has become large enough to surround the shaft and if the crank arm be then removed so that what was the crank pin is fastened directly on the shaft, an

FIG. 103.

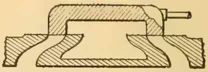


FIG. 104.—Slide Valve without Lap.

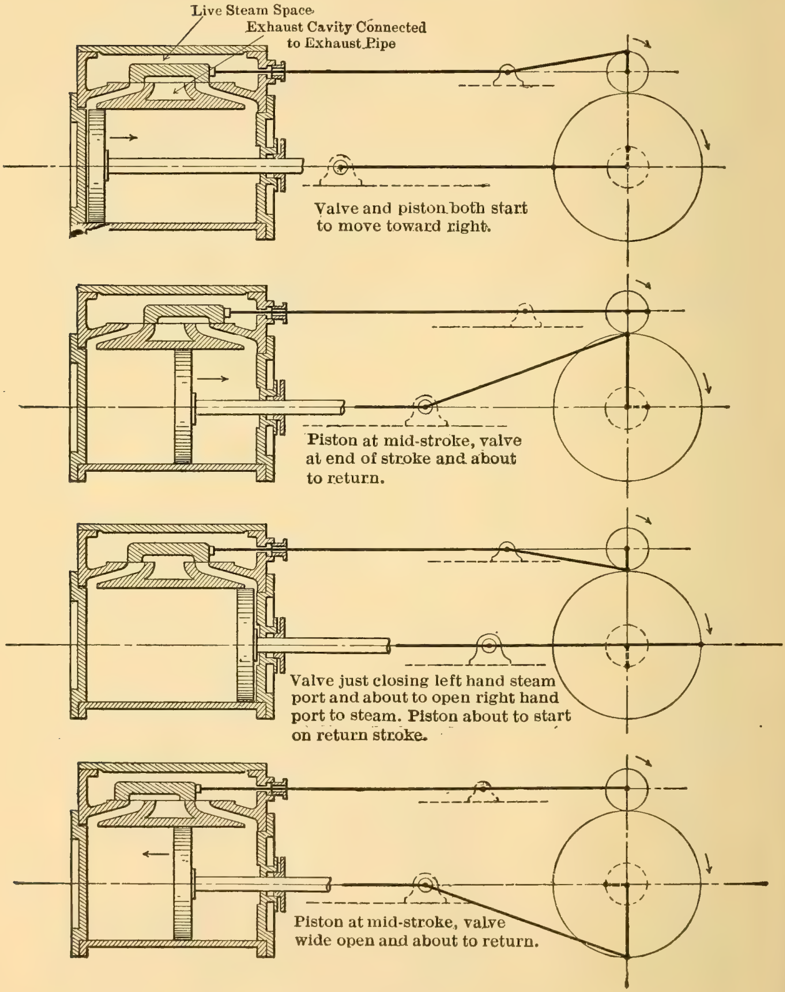


FIG. 105.

eccentric results. It is the exact equivalent of the original crank; its center, which is the center of the crank pin, revolves about the center line of the shaft in a circle with a radius a just as in the original mechanism.

The eccentric makes it possible to place a short crank (short arm) upon a large diameter shaft without having to cut the shaft away as shown in Fig. 103, and it is therefore very useful for driving valves.

78. Steam Lap. The simplest possible form of D-slide valve would just reach the outer edges of the ports when in its central position as shown in Fig. 104. The crank driving it (that is the crank equivalent to the eccentric which would probably be used in a real case) would have to be located 90° ahead of the engine crank in the direction of rotation, as can easily be seen by consulting Fig. 105, which illustrates the mechanism in various critical positions. The illustration shows that such a valve would give full stroke admission, thus producing a rectangular cycle which has already been shown to be very inefficient as a means of obtaining work from the heat used in forming steam.

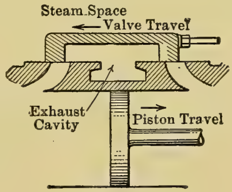


Fig. 106.

If cut-off is to occur before the end of the stroke, the edge of the valve must return and close the port before the piston reaches the end of its stroke. But since the crank mechanism does not permit the valve to remain stationary in any one position, such early cut-off could only occur if the valve over-traveled, as shown in Fig. 106, and this would unfortunately result in connecting the working end of the cylinder to exhaust and in admitting steam to the other side of the piston at such a time as to oppose the piston's motion. The solution of the difficulty lies in making the valve longer, so that when in its central position it overlaps the outer edges of the ports as shown in Fig. 107.

The amount of overlap of the outer edge is called the **outside lap**, and when steam is admitted by the outer edges of the valve, as in the case under discussion, it is also called the **steam lap**.

With such an arrangement the valve must be drawn out of its central position by the amount of the lap when the piston is at the end of its stroke as shown by *a* in Fig. 108 in order that steam may be admitted just as the piston starts to move. It follows that the crank driving the valve must be more than 90° ahead of the engine crank and that it must be ahead by the angle required to move the valve a distance equal to the outside lap. This angle, represented in the figure by α, is called the **lap angle**.

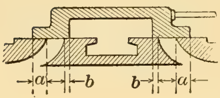


FIG. 107.—Steam and Exhaust Lap.

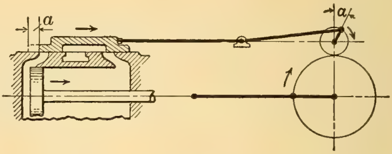


FIG. 108.—Lap *a* and Lap Angle α.

79. Lead. In real engines it is further desirable to start the admission of steam just before the piston arrives at the end of its stroke. This assists in bringing the moving parts to rest, raises the pressure in the clearance to full value before the piston starts, and gives a wider opening through which the steam can flow during the early part of the stroke, thus reducing wiredrawing and loss of area at the top of the diagram. If the valve is to open before the piston reaches the end of its stroke, the crank driving it must be shifted still further ahead of the engine crank. It must be shifted ahead by an angle which will draw the valve through the distance which will give the desired opening of valve with the piston at the end of its stroke as shown by *b* in Fig. 109. The angle required, indicated

by β, is known as the **angle of lead**, and the width of the steam opening with engine crank on dead center, i.e., the distance b, is known as the **lead**. The lead varies from less than $\frac{1}{16}$ in. on small engines and with low speeds up to over $\frac{1}{8}$ in. on large engines and with very high speeds.

80. Angle of Advance. The eccentric or valve-operating crank must be ahead of the engine crank by an angle equal to $90°+$angle of lap $\alpha+$angle of lead β, as can be seen

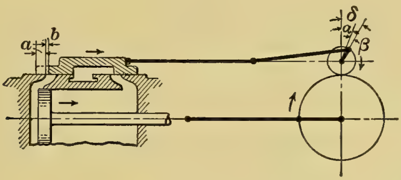


FIG. 109.—Lead b and Lead Angle β.

by an inspection of Fig. 109. The sum of α and β is called the **angle of advance** and will be represented by δ. This is the number of degrees in excess of 90 by which the eccentric leads the engine crank.

Fig. 109 shows that cut-off in an engine fitted with a valve having lap and lead must occur when the engine crank has turned through an angle equal to $180-2\alpha$, because the valve will then have returned to the closed position. Apparently, cut-off can be made to occur at any point in the stroke by properly choosing the value of α, but it will be discovered later that the exhaust events set a limit to increase in the value of this angle and hence do not permit of cut-off occurring earlier than a certain fraction of the stroke.

81. Exhaust Lap. Inspection of Fig. 105 will show that the simple valve without lap originally assumed will give no compression, because the cylinder end is connected to the exhaust cavity for the entire stroke. Inspection of all

the changes which have been suggested in the subsequent paragraphs will show further that if the inner edges of the valve are left in the original positions the exhaust events will be considerably distorted in the case of a valve having steam lap and lead.

This trouble may be remedied by moving the inner edges of the valve closer together, making the exhaust cavity in the valve shorter and giving *inside lap* as shown in Fig. 107 by *b*. When the inner edges of the valve control exhaust, as in the case of the valve under discussion, this inside lap is also called exhaust lap.

The length of the valve, the lap and the lead are generally chosen so as to give the desired arrangement of admission and cut-off and then the exhaust edges are so located as to give desirable release and compression. In some forms this necessitates the use of an exhaust cavity in the valve such as that shown in Fig. 110. The amount by which the edges of the valve fail to meet the inner edges of the port is spoken of as **negative inside lap.** This dimension is indicated by *c* in the figure.

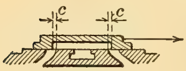


Fig. 110.

It should be noted particularly that all measurements of lap are made with the valve central on its seat and that the measurement of lead is made with the piston at the end of its stroke, i.e., with the engine crank on dead center.

82. The Bilgram Diagram. The action of all slide valves could be studied by means of drawings of the actual mechanism, as has been done in preceding paragraphs, but such a method is time and space consuming. Numerous diagrams such as the Elliptical, the Sweet, the Zeuner and the Bilgram have been developed for the purpose of simplifying and expediting such studies and, when properly understood, they are very convenient. The scope of this book does not permit a discussion of all of these diagrams

and, since the Bilgram diagram is probably the most generally applicable, attention will be confined to it.

The construction of this diagram is illustrated in Fig. 111. The point O represents the center of the engine crank shaft and the two circles drawn about this point as a center represent respectively the paths traveled by the

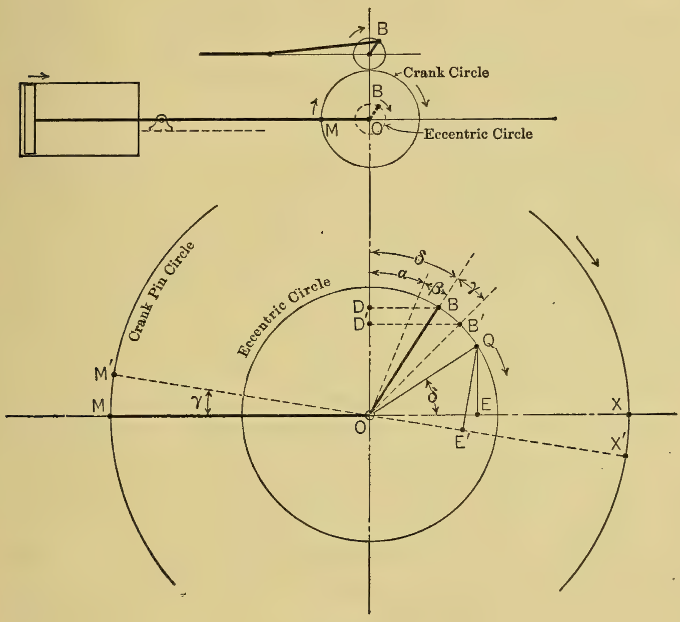


Fig. 111.

pin of the valve crank and the pin of the engine crank. These circles are drawn to any convenient scales.

The diagram is conventionally drawn in such a way that the line OM represents the head end dead center position of the crank and in all subsequent paragraphs the relative positions shown by the small sketch in Fig. 111 will be assumed. The cylinder will be assumed to the

left of the shaft and the engine will be assumed to run "over."

With the crank in position OM, the eccentric (equivalent crank) must be in the position OB, ahead of the crank by an angle $90°+\alpha+\beta=90+\delta$. The valve must then be displaced to the right of its central position by an amount represented by the distance DB, if a small correction for "angularity" of the valve connecting rod be neglected. As rotation continues, horizontal distances corresponding to this line will always give the instantaneous valve displacements. For position OB', for instance, the valve displacement will be $D'B'$.

If the angle δ is now laid off above OX, locating the point Q as shown, a perpendicular QE dropped upon OX from this point will equal in length the line DB, and will therefore show the valve displacement when the crank is in head end dead center position OM. This must be true, because the triangles QOE and BOD are similar and have the sides OQ and OB equal to the radius of the same circle.

The perpendicular QE is really a perpendicular dropped upon the extension of the line representing the crank position, and it is a general property of this diagram that a line starting at Q and perpendicular to the line representing any chosen crank position (or an extension of that line) will show by its length the displacement of the valve when the crank is in the chosen position. Thus assume the engine crank to rotate through the angle γ to the position OM'. The eccentric will have rotated to B' and the valve displacement will be represented by $D'B'$. A perpendicular drawn from Q upon OX', the extension of the crank position, gives QE' equal to $B'D'$ and hence representing the valve displacement to the same scale.

This construction drawn for different crank positions OA, OM, OM_1, OM_2, etc., is shown in Fig. 112, the dash-dot radial lines about Q representing the various values of the valve displacement. The number on each of these

lines indicates the crank position to which it corresponds. It will be seen that the displacement increases in value until the crank position OM_3 is reached, after which it decreases again.

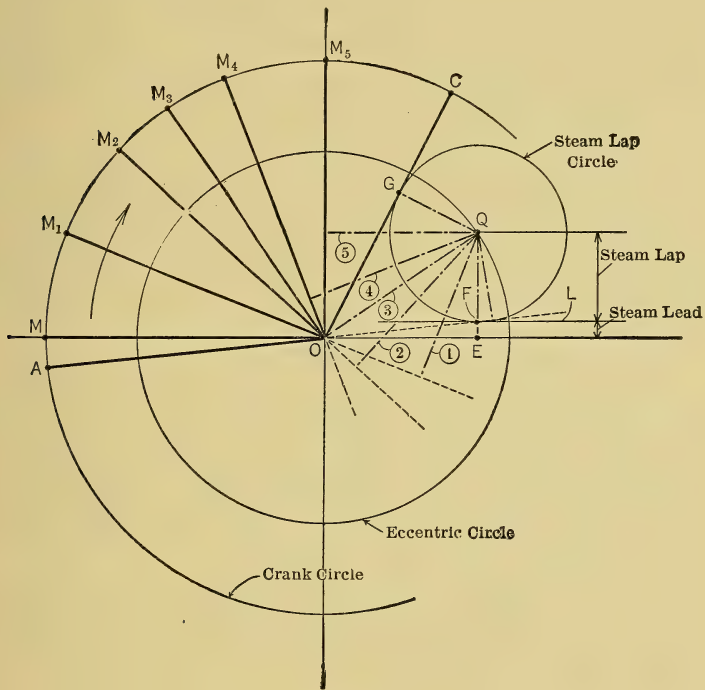


FIG. 112.

Since the opening to steam is equal to the displacement minus the lap, as shown in Fig. 109, the actual amount by which the valve is open for any crank position can be found by subtracting from the corresponding valve displacement the amount of lap possessed by the valve. For head end dead-center position, the displacement is equal to lap plus lead, and is shown by QE in Fig. 112. Subtract-

ing the lead EF, the remainder FQ gives the lap of the valve. A circle drawn about Q with radius equal to QF (or a circle drawn about Q and tangent to the line L) will cut off of the lines representing valve displacement the amount representing the part of each displacement used in overrunning the lap of the valve. The remainders, that is the parts of the lines radiating from Q in Fig. 112 which are outside of the lap circle, must then represent the amounts by which the valve port is actually open.

It will be observed that the valve is open by the amount of the lead when the crank is on dead center, position OM. The crank position for which the valve displacement is just equal to the lap, and hence at which the valve is just beginning to open, can be found by drawing a tangent through O to the lower side of the lap circle and then extending it to give the crank position OA in Fig. 112.

As the crank rotates clockwise from this position, the valve opens wider until, when position OM_3 is reached, the greatest valve opening exists. Further rotation results in partial closure of the valve and, when the crank has finally rotated into position OC, the valve has just closed, that is, cut-off has occurred, the displacement being just equal to QG, the steam lap.

Thus this diagram, as so far developed, indicates crank positions for admission and cut-off and the values of valve displacement and valve openings for all intermediate crank positions.

ILLUSTRATIVE PROBLEM

A certain valve has an external steam lap equal to $1\frac{1}{4}$ ins. The lead is $\frac{1}{16}$ in. and the throw of the eccentric is $2\frac{1}{2}$ ins. (*a*) Construct such parts of the Bilgram diagram as are necessary to indicate "head end" crank positions for admission, maximum valve opening and cut-off. (*b*) Indicate on this diagram the amount of valve opening at various crank positions between admission and cut-off. (*c*) Determine the value of the angle of advance.

Draw a circle with radius equal to the eccentric throw, $2\frac{1}{2}$ ins., using any convenient scale. This circle is designated by *abcd* in Fig. 113. Draw about the same center another circle of any convenient size. Draw in the horizontal diameter *ac* and extend as shown. On the right-hand side of the circle draw the line *ef*,

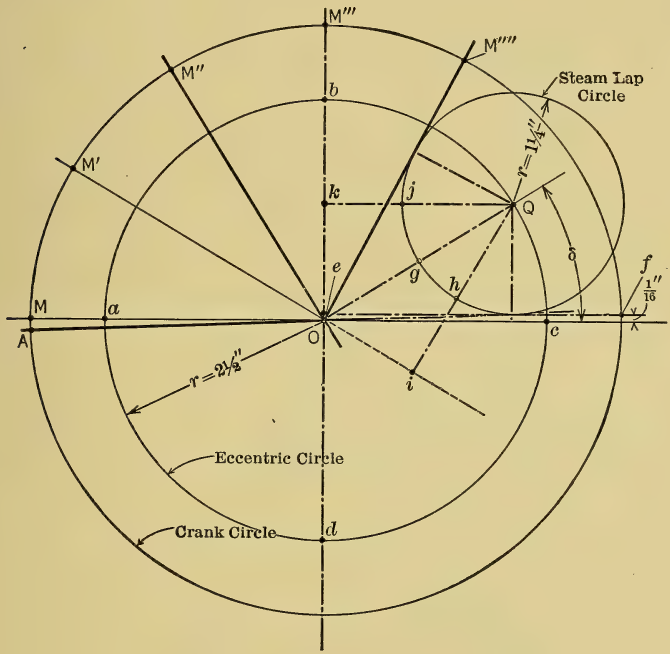
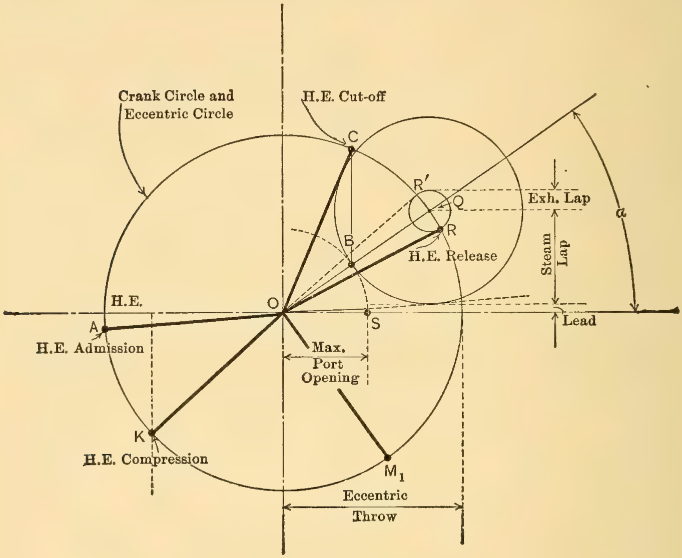


FIG. 113.

parallel to the horizontal axis and a distance above it equal to the lead, $\frac{1}{16}$ in., to the same scale as that chosen for eccentric circle. The steam lap circle must have its center Q on the upper right-hand quadrant of the eccentric circle, and it must be tangent to the line *ef*. Its radius must equal the steam lap, $1\frac{1}{4}$ in. to scale. Therefore, with compass points set the proper distance apart, find the center Q, about which a $1\frac{1}{4}$-in. radius circle will just be tangent to the line *ef*, and draw the steam lap circle.

The crank position at admission is found by drawing the line AO so that, if extended, it is tangent to the lower side of the steam lap circle.

The crank position at cut-off is found by drawing the line



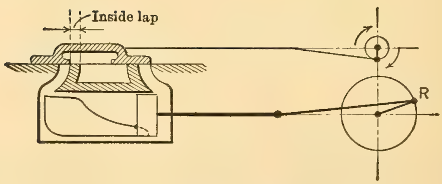


FIG. 114.

$M''''O$ in such position that it is tangent to the upper part of the steam lap circle.

The crank position for maximum valve opening is found by drawing the line $M''O$ in such position that a line through QO will be perpendicular to it. The amount of valve opening at this

crank position is shown by the length of the part of this perpendicular line outside of the steam lap circle. i.e., the distance Og interpreted according to the scale chosen for eccentric and steam lap circles.

When the crank is in position $M'O$, the length of hi, interpreted to scale, gives the amount by which the valve is open to steam.

When the crank is in position $M'''O$, the length of jk, interpreted to scale, gives the amount by which the valve is open to steam.

The angle indicated by δ is equal to the angle of advance because of the property upon which the construction of this diagram is based.

83. Exhaust and Compression. The exhaust edge events can be shown on the Bilgram diagram by a method similar to that used for the steam edge events. The direction in which valve displacements occur are indicated in the upper part of Fig. 114 in which the crank and eccentric circles have been drawn to such scales that they coincide. Inspection of the small sketch in the lower part of the figure will show that head end release must occur when the valve has traveled a distance equal to the inside lap to the left of its central position. A crank position OR drawn tangent to the lower part of a circle about Q with radius equal to the inside lap will, therefore, be the crank position at release. Clockwise rotation from this position will result in a wider opening to exhaust until position OM_1 is reached, after which the valve will begin to close. Final closure will occur when the crank reaches position OK, the extension of which is tangent to the top of the exhaust lap circle. At that time the valve will have returned (moving from left to right) and will still have to move a distance equal to the exhaust lap before attaining a central position.

ILLUSTRATIVE PROBLEM

Given the exhaust lap of a D-slide valve equal to $\frac{3}{8}$ in.; the steam lap $1\frac{1}{4}$ ins.; the throw of the eccentric, 2 ins.; and the lead $\frac{1}{8}$ in. Find the angle of advance, the maximum port opening to steam and to exhaust, and the crank positions of cut-off, release, compression and admission for the head-end of the cylinder.

Draw the eccentric (and crank) circle with a radius equal to 2 ins., and draw the horizontal diameter as in Fig. 115.

Draw a horizontal line in the upper right-hand quadrant at a distance of $\frac{1}{8}+1\frac{1}{4}$ ins. above the horizontal diameter. Locate the point Q at intersection.

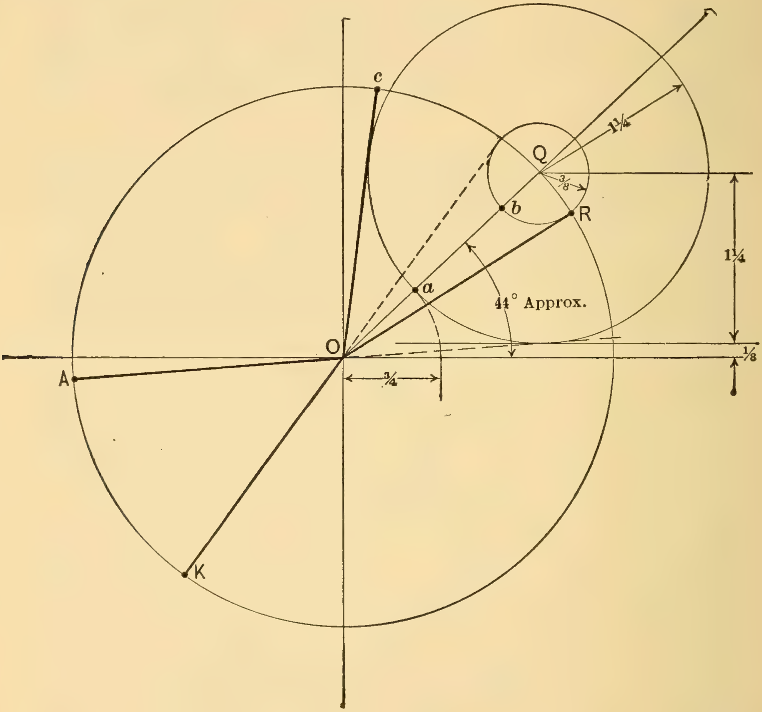


FIG. 115.

Draw the steam lap circle with a radius $1\frac{1}{4}$ in. and the exhaust lap circle with a radius $\frac{3}{8}$ in.

The angle of advance is the angle between OQ and the horizontal.

The maximum opening to steam is given by the distance $Oa=\frac{3}{4}$ in. The maximum opening to exhaust is given by the distance $Ob=1\frac{5}{8}$ in.

The crank positions shown are obtained by drawing lines

tangent to the lap circles. *A* represents admission; *C*, cut-off; *R*, release, and *K*, beginning of compression.

The piston positions at the times of these events are given to reduced scale by vertical projection.

84. Diagram for Both Cylinder Ends. The complete diagram for the head end of cylinder is shown in Fig. 114 with all critical crank positions marked. The positions for the crank end of the cylinder can be found in a similar way by constructing a diagram in which the point *Q* and the lap circles are located in the opposite quadrant. The resulting

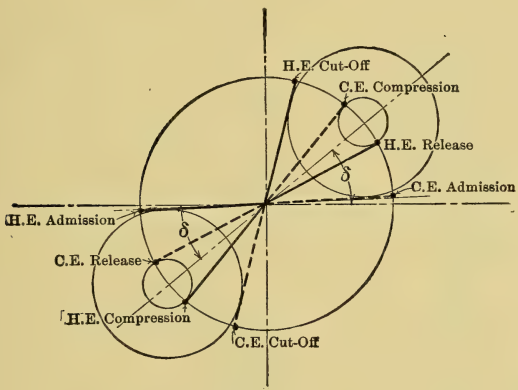


FIG. 116.

diagram for both cylinder ends, with laps the same for both ends of the valve, is given in Fig. 116.

85. Piston Positions. The valve events might be studied entirely in conjunction with crank-pin positions, but it is more convenient and customary to consider them in connection with piston positions. Piston positions corresponding to different crank-pin positions could be found by drawing the mechanism to scale for each different position as shown in Fig. 117 for piston positions 1 and 2.

It is obvious that this would involve a great deal of work and that, if drawn to large scale, it would consume a great

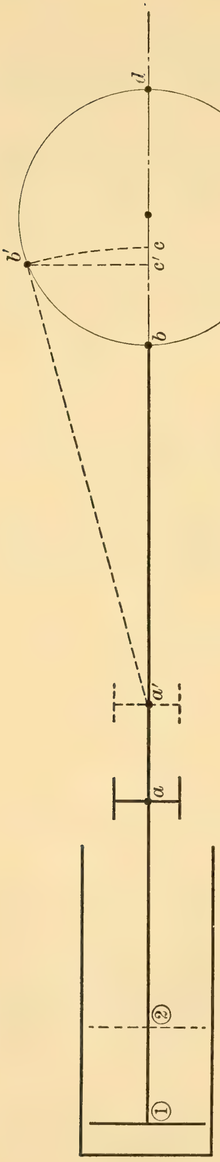


Fig. 117.

deal of space. Further, it is convenient to be able to locate relative piston positions on the line which serves as the horizontal diameter of the crank circle of the Bilgram diagram.

The method used depends upon the fact that the motion of the crosshead is exactly the same as that of the piston, so that if the motion of the crosshead end of the connecting rod can be followed, it will be equivalent to following the motion of the piston itself. It should also be noted that the diameter of the crank circle must be equal to the stroke of the engine.

Assume now, that the point b in Fig. 117 be taken to represent the position of the piston when it is really in position 1. When the piston has moved to position 2, the crosshead will have moved from a to a' and the crank pin from b to b'. If with a' as a center the connecting rod be swung down to the horizontal its right-hand end will arrive at the point c. The distance bc must then represent the distance that crosshead (and piston) have moved from dead-center position because ab and

$a'c$ both represent the length of the connecting rod and c must therefore be as far to the right of b as a' is to the right of a. The point c may therefore be taken to represent piston position when the connecting rod is in the position $a'b'$.

In general, if the horizontal diameter of the crank circle be taken to represent the stroke of the engine, the piston position corresponding to any crank position can be found by taking a radius equal to the connecting-rod length (to the same scale as the circle) and striking an arc from the

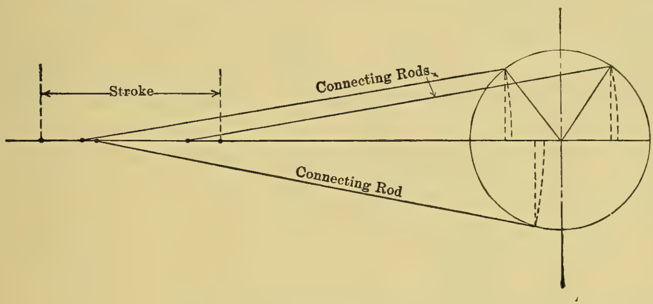


FIG. 118.

crank-pin position, using a center on the horizontal line on the cylinder side of the crank circle.

An approximate method is also used for finding the piston position. Instead of projecting down from the crank-pin position with an arc, such as $b'c$ in Fig. 117, a vertical line through the crank-pin position is used. Such a line would give c' as the piston position when c is really correct. This method would give accurate results with a connecting rod of infinite length. For ordinary lengths of rod, however, the results are far from correct. The error is said to be due to the **angularity of the connecting rod.**

The effect of the angularity of the connecting rod is shown in Fig. 118 for different positions. On the outstroke the piston is always farther ahead than the rectilinear pro-

jection would indicate. On the return stroke the piston is always behind the position indicated by rectilinear projection.

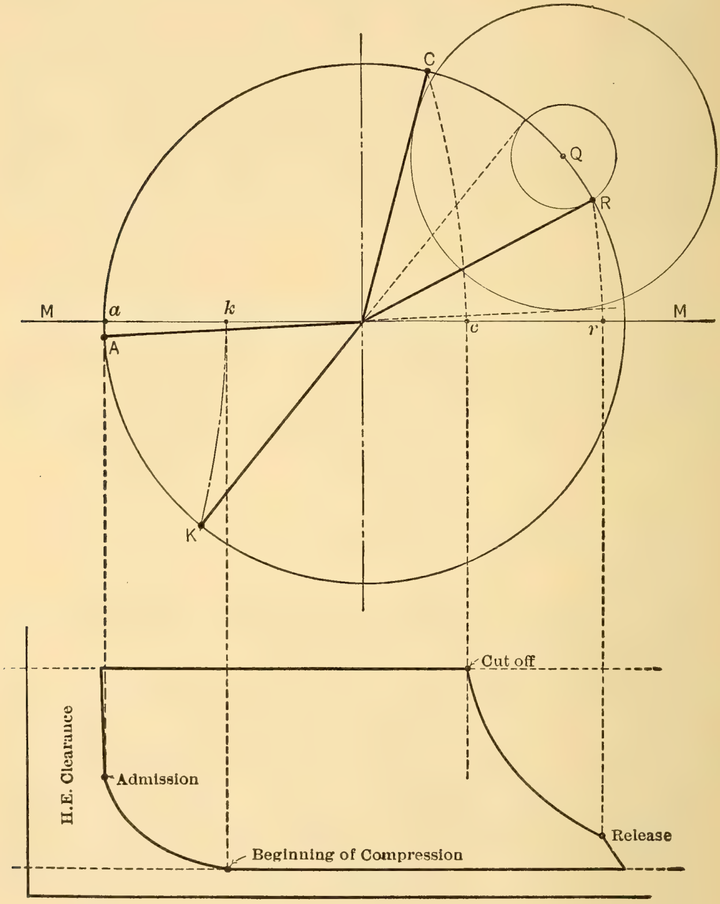


FIG. 119.

86. Indicator Diagram from Bilgram Diagram. Since the piston positions corresponding to different crank posi-

tions can be determined, it is a comparatively simple matter to construct the indicator diagram which theoretically would be given by an engine fitted with a valve of certain dimensions. It is necessary to assume the upper and lower pressure and also to assume the form of the expansion and compression curves. These are generally taken as rectangular hyperbolas.

The method of constructing an indicator diagram from the Bilgram diagram is shown in Fig. 119. The crank-pin positions for admission (A), cut-off (C), release (R) and beginning of compression (K) are first found. These pin positions are then projected to the horizontal diameter by means of arcs with radius equal to the connecting-rod length and with centers on the line MM produced to the left. The intersections a, c, r and k indicate the piston positions at which the corresponding events occur. These are then projected vertically downward to intersect the proper pressure lines and the card is drawn through the intersections.

Diagrams constructed in the same way, but for both head and crank ends, are given in Fig. 120. A symmetrical valve was assumed, that is, one built exactly alike on head and crank ends. The diagrams show that such a valve cannot give the same results for both cylinder ends because of the effect of the angularity of the connecting rod. It is most evident in the case of cut-off. The cut-off in this case occurs just before three-quarter stroke for the head end and just after half stroke for the crank end of the cylinder. All other events are distorted in the same way, but the actual lengths of the variations are not as great as in the case of the cut-offs and therefore the distortion is not as obvious.

The effect of the angularity of the connecting rod upon the diagrams can be remembered easily if it is noted that all valve events occur later with respect to piston position on the outstroke and earlier on the instroke than they would with a connecting rod of infinite length.

It is possible to "equalize" the cut-offs, that is, make them occur at the same fraction of the stroke by using unequal steam laps at opposite ends of the valve, but this will result in still further distortion of admissions, as can be seen by constructing a Bilgram diagram for this case. Similarly, the compressions can be equalized by the use of

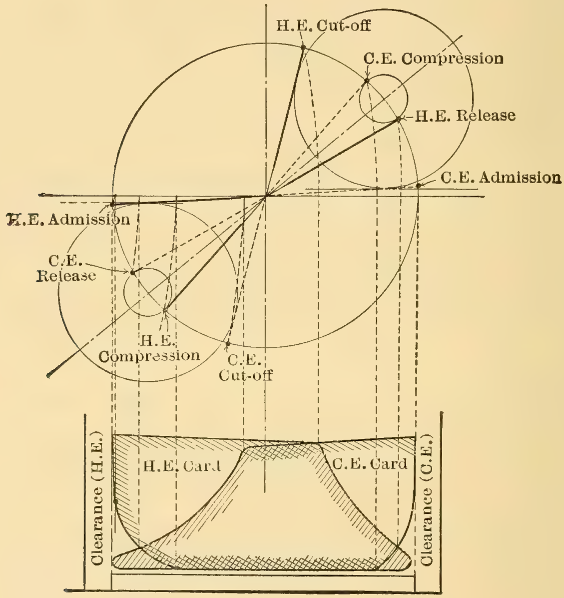


Fig. 120.

unequal exhaust laps, but this results in distortion of the release events.

Various linkages have been developed which are so arranged that they distort the motion of the valve to just the extent necessary to counterbalance the effects of the angularity of the connecting rod. The scope of this book does not, however, permit a discussion of such valve gears.

87. Limitations of the D-slide Valve. The simple valve discussed in the preceding paragraphs has numerous limitations and is therefore only used on small and cheap engines, or in cases where economy in the use of steam is not essential. This valve, when used with steam entering over the outside edges as previously considered, is pressed to its seat by the live steam acting over its entire upper surface. This pressure is practically unbalanced, as the greater part of the lower surface of the valve is subjected to the low pressure of the steam being exhausted. As a result the friction to be overcome in moving the valve is very great and there is an appreciable loss from this source.

Further, the shape of the valve makes necessary the use of long ports which form part of the cylinder clearance and which are alternately exposed to live and to exhaust steam with results previously discussed. These ports can be decreased in length by increasing the length of the valve, but this in turn increases the area exposed to high pressure and hence increases the friction loss.

It can be shown by means of the Bilgram diagram that, if a cut-off earlier than about $\frac{5}{8}$ stroke is desired, the angle of advance, the amount of steam lap and the size of the eccentric must all be made very great. This results not only in large friction losses, but also in very early release and compression, because of the great angle of advance. As a result, slide valves of the simple D type are seldom used when a cut-off earlier than $\frac{1}{2}$ to $\frac{5}{8}$ stroke is desired. It should be remembered in this connection that the simple engine generally gives its best economy with a cut-off of about $\frac{1}{4}$ stroke.

The drawing of lines representing the opening of the valve to steam as in Fig. 112 will show that this simple valve is further handicapped by the very slow opening and closing of the steam ports, causing a great amount of wire drawing with a corresponding loss of diagram area. In order to get an adequate opening to steam the valve

must also be given a great displacement and, since this occurs under great pressure, it results in great friction loss.

The unbalanced feature can practically be overcome by rolling up the valve and ports about an axis parallel to the length of the cylinder. This gives what is known as a **piston valve,** shown diagrammatically in Fig. 121

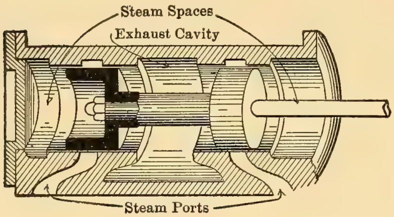


Fig. 121.—Piston Valve.

It can also be partially overcome by using a balance plate or ring of some kind between the top of the valve and the inside of the steam-chest cover, so arranged that live steam is excluded from the greater part of the upper surface of the valve. Valves of this type are generally called **balanced slide valves** and are used on many high- and medium-speed engines.

The valve travel required for obtaining a given opening

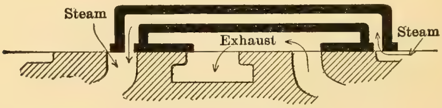


Fig. 122.—Allen Double Ported Valve.

can be decreased and the rate of opening and closing can be increased by the use of multiported constructions. These are so arranged that two or more ports open or close at the same time, so that the total movement required for a given opening is divided by the number of ports and the rate of opening and closing is multiplied in the same proportion. One simple type of double-ported valve is illustrated in Fig. 122.

When several ports are used the valve often becomes

a rectangular frame crossed by a number of bars and is known as a **gridiron valve,** because of its appearance. Such valves are often combined with balance plates and give very satisfactory results.

A number of designs of slide valves have been developed for the purpose of making cut-off independent of the other events. Many of these use a separate cut-off valve which either controls the steam supply to the main valve or else rides on the main valve and controls cut-off by covering ports in that valve. Devices of the latter type are called **riding cut-off valves.** They are either driven by separate eccentrics, or by linkage from the eccentric controlling the main valve, the linkage being so arranged as to give the proper relative motion between main and auxiliary valves. In such designs the main valve is proportioned so as to give the desired admission, release and compression and the cut-off is then taken care of by proper adjustment of the cut-off valve.

88. Reversing Engines. It was shown in one of the early paragraphs of this chapter that the eccentric must be set $90°+$angle of advance ahead of the crank, ahead meaning in the direction of rotation. To cause the engine to revolve in the opposite direction, that is, to "reverse" the engine, it is therefore only necessary to shift the relative positions of eccentric and crank so that the eccentric leads the crank by $90°+\delta$ in the new direction of rotation. This corresponds to shifting ahead (in first direction of rotation) through an angle equal to $180-2\delta$ or shifting backward through an angle equal to $180+2\delta$, as can be seen by inspection of Fig. 109.

In practice it is generally more convenient to use two eccentrics, one set properly for rotation in one direction and the other set properly for rotation in the opposite direction. This arrangement is shown diagrammatically in Fig. 123. This figure is drawn for a vertical engine and in such position that the engine is on crank-end dead center.

The point P represents the position of the center of the crank pin; the point f represents the position of the equivalent crank (center of eccentric) which drives the valve for "forward," "ahead" or clockwise rotation; and the point b represents the position of the equivalent crank which drives the valve for "backing," "reverse," or counter-clockwise rotation.

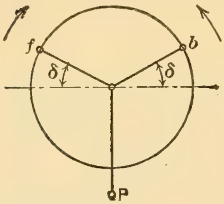


FIG. 123.

The real mechanism, in one of its numerous forms known as the Stephenson Link Gear, is shown in perspective in Fig. 124. The forward eccentric corresponds to f of Fig. 123 and the backing eccentric corresponds to b of that figure. The eccentric rods are fastened to opposite ends of a curved "link" and move the valve through a "link block" fastened to the end of the valve stem. In the position shown in the figure the link is in such position that the forward eccentric operates practically directly on the valve stem so that the valve motion is practically entirely governed by that eccentric. If the reverse shaft were to be rotated clockwise into the backing position, the "suspension rods" would pull the link over until the eccentric rod of the backing eccentric was directly under the valve stem. Under such conditions the valve motion would be controlled almost entirely by the backing

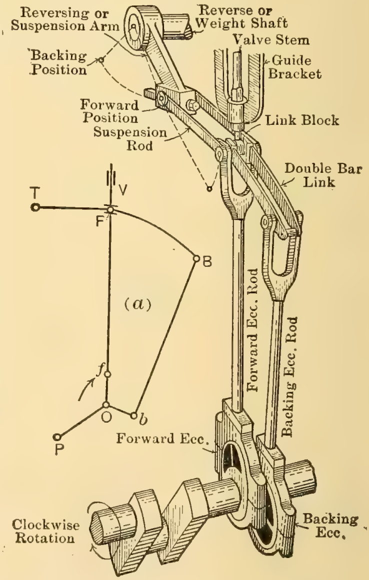


FIG. 124.—Stephenson Link Gear.

eccentric and the engine shaft would rotate counter-clockwise.

If the mechanism were so set that the link block occupied a position on the link between the ends of the two eccentric rods, the valve motion would be controlled by both eccentrics and would be a compromise between the motions given by either eccentric separately. It is characteristic of this gear that the cut-off is latest when either one or the other eccentric is fully "in gear" and that it becomes earlier as the link block approaches the center of the link. With the link block in the center of the link the valve does not open at all, i.e., the cut-off occurs at zero stroke.

There are numerous other forms of link gears, the best known being the Gooch, the Allan and the Porter-Allen. There are also numerous reversing mechanisms known as radial gears in which the motion of the valve is controlled by means of a "radius rod" which can be set to give the desired valve motion. The valve motion is obtained indirectly through the radius rod from an eccentric, from the crank, or from the connecting rod. The limits of this book do not permit a detailed discussion of these forms.

89. Valve Setting. From what has preceded it will be evident that it is not only necessary that a valve and its seat and driving mechanism be correctly designed, but also that the various parts must be correctly connected up in order that the valve may move in its proper phase relation with respect to the piston.

Adjusting the mechanism in such a way that the proper phase relations are obtained is known as **setting the valve.** This can be done with fair accuracy by a simple study of the mechanism in various positions, as will be shown below, but it is always advisable to check the setting by means of indicator diagrams taken after the setting is completed. Such diagrams will often show errors of such character or size that they cannot be determined by measurement on an engine which is not operating.

Before beginning operations it is always advisable to go over the entire engine carefully and to eliminate excessive lost motion at all pins and bearings in order that the relative positions of parts obtained while setting the valve may approximate those which will be obtained when the engine is in operation. The effect of lost motion will be appreciated after a study of Fig. 125. Assume that all parts of the mechanism are tight except the crank-pin end of the connecting rod as shown. If, for instance, the engine is rotated by hand by turning the fly-wheel, the crank will pull the piston mechanism and the piston will be drawn into the position shown in the upper half of the figure when the crank has turned through an angle α. On the other hand, when the engine is operating under steam, the piston will push the crank pin around and will occupy a position such as that shown in the lower half of the figure when the crank has been turned through the same angle α. Obviously, the piston can occupy two very different positions for the same crank position, and a valve setting based upon the conditions shown in the upper part of the figure might be

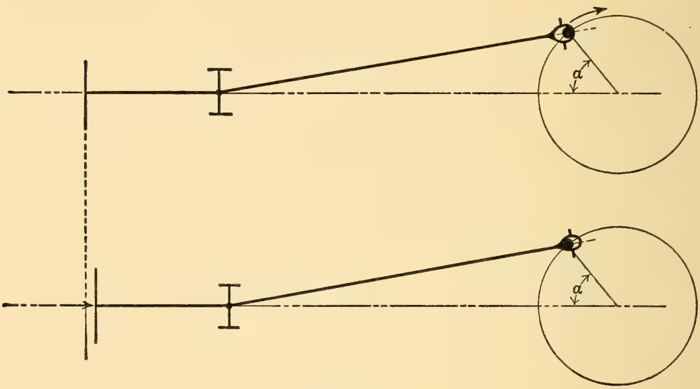


Fig. 125.

very incorrect when used under the conditions shown in the lower part of the figure.

Lost motion in any part of the mechanism can produce analogous results and it is therefore necessary to remove as much of it as possible before attempting to set the valve. It is practically impossible to eliminate all lost motion, as there must be sufficient clearance at all bearing surfaces to accommodate a film of oil, and this alone would make necessary the taking of indicator diagrams for the checking of valve settings, even if it were possible to set perfectly by measurement for stationary conditions.

In general, there are two adjustments which can be made in setting a plain slide valve. The length of the valve stem or eccentric rod can be changed and the eccentric can be shifted around the shaft. It is necessary to understand the effects of each of these adjustments.

Changing the length of the valve stem is equivalent to shifting the valve upon its seat without moving the engine as shown in Fig. 126. In this figure the valve is shown in its central position by full lines. The lap is the same at both ends. If, now, the valve is worked to the right upon its stem by adjustment of the nuts shown, until it reaches the dotted position, the head-end lap will have been decreased and the crank-end lap will have been increased by the same amount. This would make admission earlier and cut-off later for the head end and admission later and cut-off earlier for the crank end. Obviously, the effects of changing the length of the valve stem are opposite for the two ends of the cylinder.

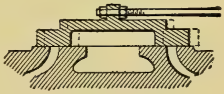


FIG. 126.

Shifting the eccentric about the shaft simply changes the time relation between valve motion and piston motion; it does not alter the valve motion itself. If difficulty is experienced in realizing the truth of this statement, it is only necessary to draw several Bilgram diagrams for the

same valve, but with different angles of advance, and then to construct indicator diagrams for both cylinder ends in every case. It will be discovered that shifting the eccentric ahead in the direction of rotation, for instance, will make all events occur earlier with respect to piston position for both ends of the cylinder.

In setting a plain slide valve which is built symmetrical about a central axis, i.e., same inside and outside lap at each end, it is first necessary to adjust the length of the valve stem. This may be done by removing the steam-chest cover so as to expose the valve and then rotating the engine slowly by hand and observing the distance traveled by the valve on each side of its central position. This is conveniently done by observing the distance between the outer edge of the steam port and the outer edge of the valve when the valve is fully open at each end. If the valve travels further toward the head end than it does toward the crank end, with reference to the port edges, the valve stem must be shortened; if it travels further toward the crank end the stem must be lengthened.

In making these adjustments it is advisable to turn the engine only in the direction in which it is going to rotate, so that any lost motion in the valve mechanism will have approximately the same effect as when the engine is operating.

When the length of the valve stem is correctly adjusted, the eccentric must be so set on the shaft as to give the proper angle of advance. This is commonly done by shifting it about the shaft until the proper value for the steam lead has been obtained. In order to determine the value of the lead it is necessary to be able to set the engine on each dead center. This can be done approximately by turning the engine until the crosshead has come to either end of its stroke, but it will be found by trial that the fly-wheel and shaft can be turned through a very large angle at each end of the stroke without causing an appreciable motion of the crosshead,

so that this method is not very satisfactory for the purpose of adjusting the eccentric. It is customary, therefore, to work in such a way as to give a more accurate determination of shaft and crank positions for dead center.

The engine is rotated until the crosshead has been brought near one end of its stroke, as shown in Fig. 127, and a mark is then scribed across the crosshead and guide as at *ab*. An arc *xy* is then marked on the fly-wheel by means of a tram such as that shown, the end *c* being placed

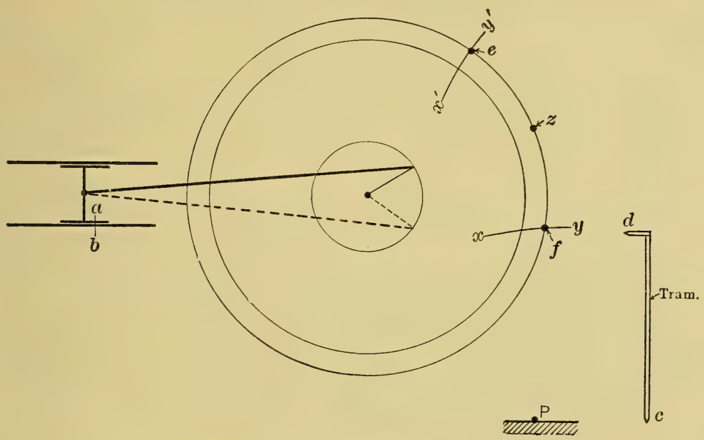
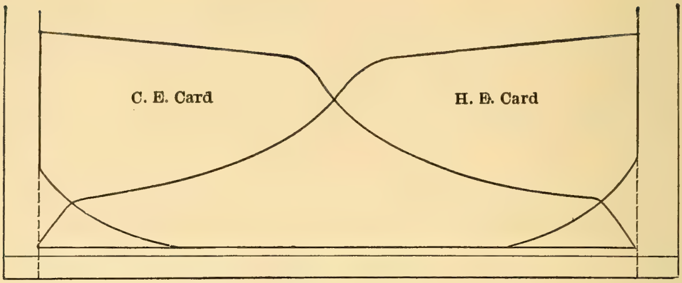
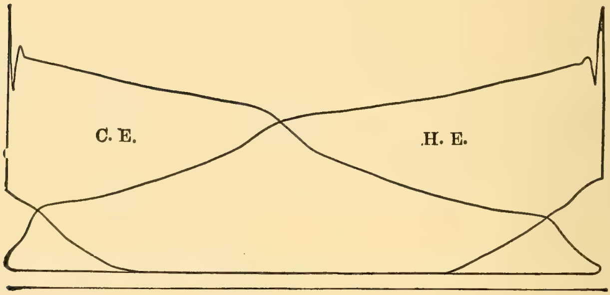


FIG. 127.

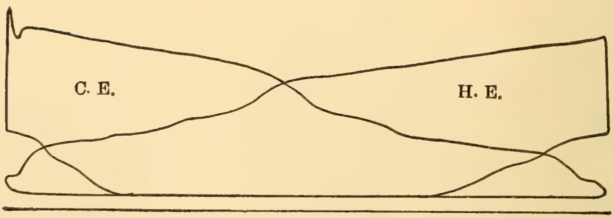
at point *P* on some solid part of foundation or floor. The engine is then rotated, clockwise in the figure, until the crosshead has reached the end of its stroke and returned to such a point that the marks on crosshead and guides again coincide, as shown by dotted positions in the figure. The arc $x'y'$ is then scribed on the fly-wheel with the tram, the end *c* again bearing on the point *P*. A point *z* is then found by bisecting the arc *ef* and when this point is brought under point *d* of the tram the crank will obviously be at crank-end dead center and the piston at the crank end



(*a*) Perfect Cards for Slide Valve Type.

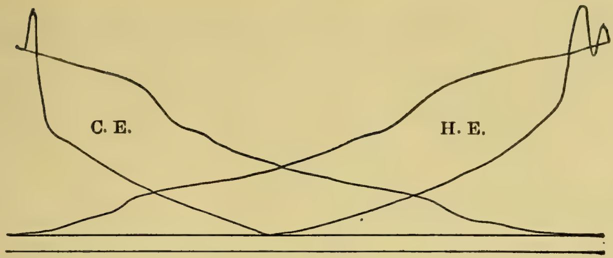


(*b*) Actual Card; Small Engine. Center Line of Valve on Center Line of Seat; Eccentric Advanced to Give Normal Lead of 0.05 inch. Engine Running Over.

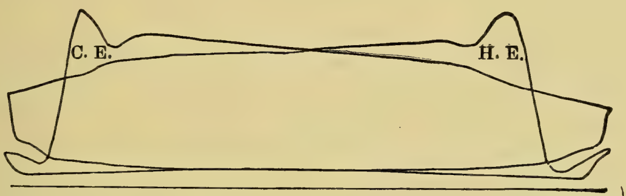


(*c*) Same Setting as (*b*) except Engine Running Under.

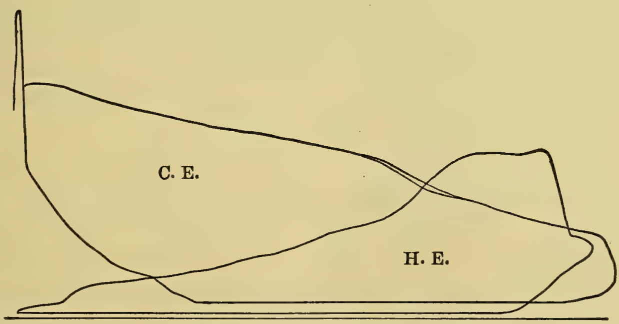
Fig. 128.



(*d*) Angular Advance of Eccentric Increased. Valve Stem Length Same as in (*b*) and (*c*). Lead 0.375 Inch.



(*e*) Angular Advance of Eccentric Decreased so as to Give Negative Lead of 0.5 Inch. Length of Valve Stem Unchanged.



(*f*) Length of Valve Stem Changed; Angle of Advance as in (*b*).

FIG. 128.

of its stroke. A point on the fly-wheel diametrically opposite to z is next found, so that when it is brought under point d of the tram the engine will be on head-end dead center.

It is probable that more accurate results are obtained by rotating the engine in a direction opposite to that in which it rotates under steam, because lost motion is then taken up in the same direction as when working, but when the whole process of valve-setting is considered it is questionable whether this is the correct direction of rotation. Opinion and practice differ in this respect. In the end, the setting should be checked by the taking of indicator diagrams, so that effects of incorrectible lost motion may be finally eliminated.

With the dead-center points found the engine is placed on, say, head-end dead center, and the eccentric shifted until the valve is open to steam by the desired lead. The eccentric is then fastened in this position and the engine turned to the opposite dead center. Because of angularity of connections and of irregularities in valve and seat dimensions, it generally will be discovered that the valve is not now open to steam by the same amount as at the other end. If it is desired that it should be, the valve can be shifted on its stem about half of the distance by which it is out and the eccentric can then be swung about the shaft to take up the remaining distance. The effect should then be checked by putting the engine on the opposite dead center.

Valves may be set for equal leads as above, or for equal cut-offs or for any sort of a compromise desired. In any case the procedure is about the same. The length of the valve stem is adjusted, then the eccentric position is adjusted, and then refinements are effected by small changes of both adjustments. Remember always, that changing the length of the valve stem changes events at opposite cylinder ends in opposite directions, while shifting the eccentric changes all events in the same direction.

The effects of various adjustments are shown by the

indicator diagrams given in Fig. 128. These diagrams were taken from a small, slide-valve engine and serve very well to show the way in which the indicator discloses poor adjustments.

PROBLEMS

1. Given: angle of advance, 30°; throw of eccentric, $1\frac{1}{2}$ ins.; lead, $\frac{1}{16}$ in.; maximum exhaust-port opening, $1\frac{1}{4}$ in.; find the steam lap, maximum opening to live steam, and the exhaust lap.

2. Given: steam lap of $\frac{7}{8}$ in.; lead of $\frac{1}{16}$ in.; exhaust lap of $\frac{3}{8}$ in.; and the angle of advance equal to 30°. Find the valve travel ($=2\times$throw of eccentric) and maximum port opening to steam and to exhaust.

3. An engine has an eccentric throw of $1\frac{3}{4}$ ins.; a steam lap of $\frac{3}{4}$ in.; and a lead of $\frac{1}{16}$ in. Compression begins at $\frac{7}{8}$ of the return stroke. Assume a connecting rod of infinite length and find the angle of advance, the exhaust lap, and the maximum port openings to steam and to exhaust.

4. Given: valve travel, 3 ins.; steam lap, $\frac{5}{8}$ in.; exhaust lap, $\frac{1}{4}$ in.; and lead, $\frac{1}{8}$ in.; find maximum port opening, angle of advance, and piston positions at cut-off, release, compression, and admission for both ends of cylinder, with the length of the connecting rod equal to $4\frac{1}{2}$ times the length of the crank.

5. It is required to build an engine having a steam-port opening of $\frac{3}{4}$ in., a lead of $\frac{1}{16}$ in., and a connecting rod four times the length of the crank. Cut-off must occur at $\frac{3}{4}$ stroke and release at 95% of the stroke. Find the inside and outside lap, the throw of the eccentric and the fraction of stroke completed by the beginning of compression.

CHAPTER XI

CORLISS AND OTHER HIGH-EFFICIENCY ENGINES

90. The Trip-cut-off Corliss Engine. The slide valve has certain limitations which can be partly, but never wholly, overcome. In most slide-valve gears, for instance, the various events occur more slowly than is desirable, and this is particularly true of cut-off. Ideal valves would open suddenly to full opening when necessary and would close as suddenly at the proper time, and such action would give minimum throttling loss and rounding of corners of the diagram. Engines fitted with such ideal valves would therefore give indicator diagrams with maximum work area as shown by the dotted lines in Fig. 129, the full lines indicating the type of diagram obtained with the ordinary slide valve.

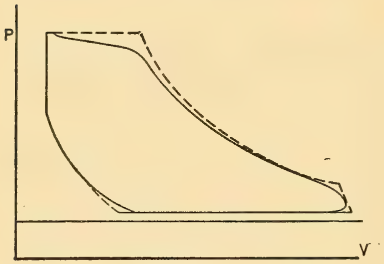


Fig. 129.

Again, the simpler forms of slide valve involve the use of long ports connecting with the clearance space within the cylinder, thus adding greatly to the clearance surface exposed and to the cylinder condensation. These ports serve for both admission and exhaust, and their walls are therefore periodically cooled by the exhaust steam with the result that excessive condensation occurs during admission.

Many attempts have been made to devise valve gears which should not be subject to the limitations of the

simple slide valve. Some of these have resulted in the development of the more complicated slide valves described in the last chapter, but such designs generally leave much to be desired. One of the earliest and most successful solutions was made by Corliss, who developed what is known as the trip-cut-off Corliss gear.

The long combined steam and exhaust ports are eliminated by the use of four valves, two for steam and two for exhaust. These are rocking valves and are located top and bottom, at the extreme ends of the cylinder, with their longitudinal axes perpendicular to those of the cylinder, as shown in Figs. 48, 49, and 50. The exhaust valves are located below so as to drain out water of condensation. Details of valves of this type are shown in Fig. 130.

These valves may each be regarded as an elementary slide valve which has a cylindrical instead of a flat face, and which is oscillated about a center near the face instead of being reciprocated, i.e., oscillated about a center at an infinite distance.

The valves are operated as shown in Fig. 131 by short links from a wrist-plate pivoted on the side of the cylinder and rocked back and forth about its center by means of an eccentric operating through the linkage indicated. The locations of the various pins and the lengths of the various links are so chosen that the valves travel at high velocity when opening and closing, that they open very wide, and that they close only far enough to prevent leakage and then remain practically stationary until about to open again. Throttling losses are thus decreased and wear caused by useless motion after closure is minimized.

The opening of the admission valves in this gear is effected positively by the linkage already explained, but they are closed differently. For opening, the steam link rotates the bell crank *B* in Fig. 132 and thus raises the latch *C*. The hook on the end of one of the arms of this latch engages the steam arm which is fastened on the end

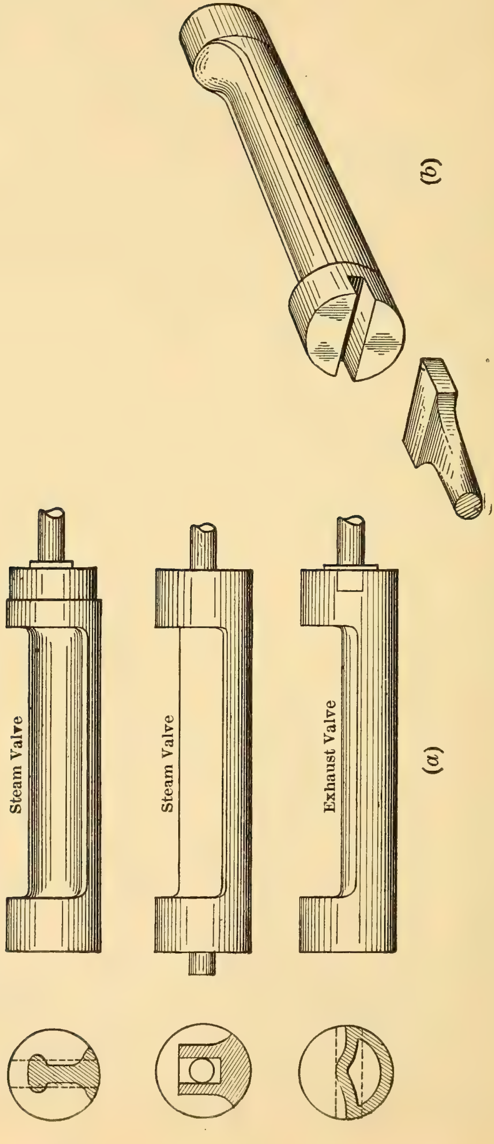


FIG. 130.—Details of Corliss Valves.

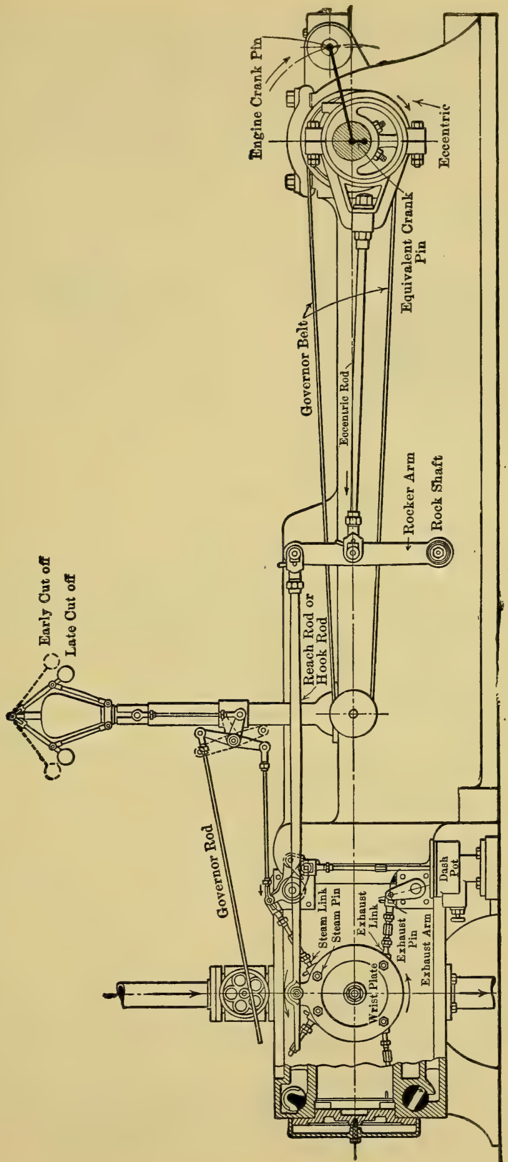


FIG. 131.—Corliss Engine Gear.

of a rod which is slotted into the end of the valve. The valve is thus drawn further open as the wrist plate revolves, until the tripping end D of the latch strikes the cam indicated by E. This throws the hook out of engagement and thus disconnects the valve from the driving mechanism. The

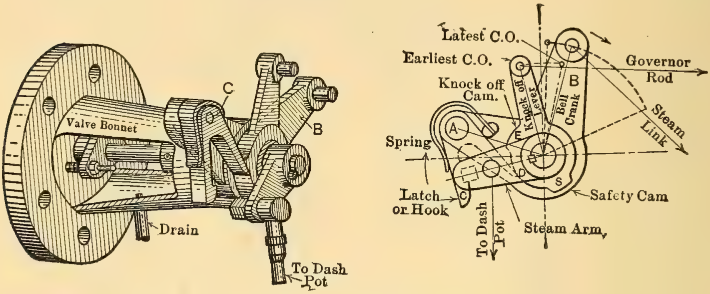


FIG. 132.—Details of Corliss Trip-Cut-off Gear.

valve is closed by the action of a dash pot, one form of which is shown in Fig. 131. As the steam arm rises during the opening of the valve it draws up the plunger or piston of the dash pot, leaving a partial vacuum beneath it, and, when the

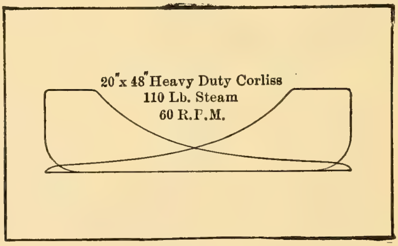


FIG. 133.

valve is released by unhooking of the latch, atmospheric pressure drives the plunger down and thus causes cut-off to occur. The action of a dash pot is found to be unsatisfactory when the speed of the engine exceeds about 125 R.P.M. and most Corliss engines with trip-cut-off operate

at still lower speeds. Under such circumstances the cut-off is very rapid as compared with the piston speed, and the diagram shows a comparatively sharp corner at this point. A set of diagrams obtained from a large Corliss engine operating at low rotative speed is given in Fig. 133, and it is obvious that little throttling occurs.

Because of the low speed at which these engines operate the stroke can be made long with respect to the diameter without attaining a prohibitive piston speed. The economy mentioned in Chapter VII as resulting from the use of long strokes can thus be obtained in these engines. An idea of the saving in steam effected by the partial elimination of throttling and condensation losses by means of the Corliss gear can be obtained from the curves in Fig. 134 (*a*) and (*b*), which give average performances.

The position of the cam which determines the time at which cut-off occurs is controlled by the governor of the engine. When moved in the direction taken by the steam arm it causes cut-off to occur later. Variation of the point of cut-off is used in these and in most other engines to control the amount of work done per cycle in order that the engine may make available the quantity demanded at the shaft, as will be explained in a later chapter. It is therefore desirable that the range of cut-off should be as great as possible, but it has been found very difficult to design trip-cut-off gears which will give a cut-off later than about 0.4 stroke if steam and exhaust valves are operated from the same eccentric. Later cut-off causes poor timing of the exhaust events.

This has led to the introduction of Corliss engines with two eccentrics and two wrist plates per cylinder. One set operates the steam valves and the other the exhaust valves. With this arrangement the range of cut-off is unlimited.

91. Non-detaching Corliss Gears. Because of the low speed at which trip-cut-off Corliss engines are operated,

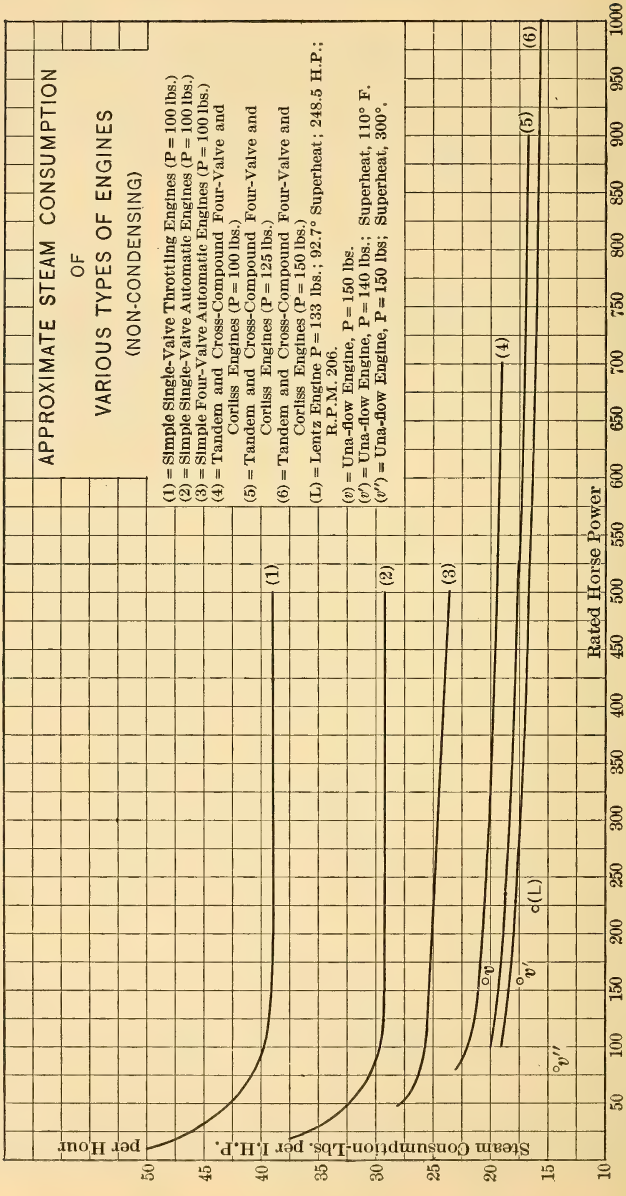


FIG. 134*a*.—Steam Consumptions of Non-Condensing Engines.

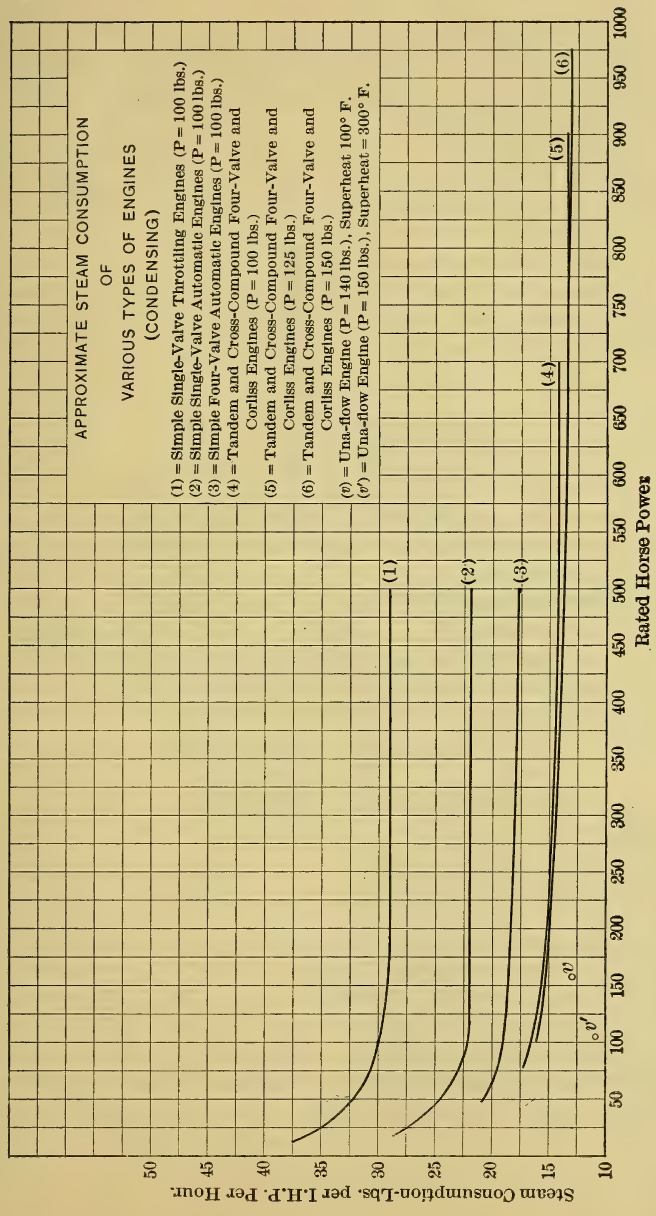


FIG. 134*b*.—Steam Consumptions of Condensing Engines.

they are necessarily large, heavy and costly and efforts have been made to design gears which shall possess the advantages of the original Corliss mechanism without the limitation as to speed.

In many models the Corliss valves are retained and are located in the ends of the cylinder as just described or in

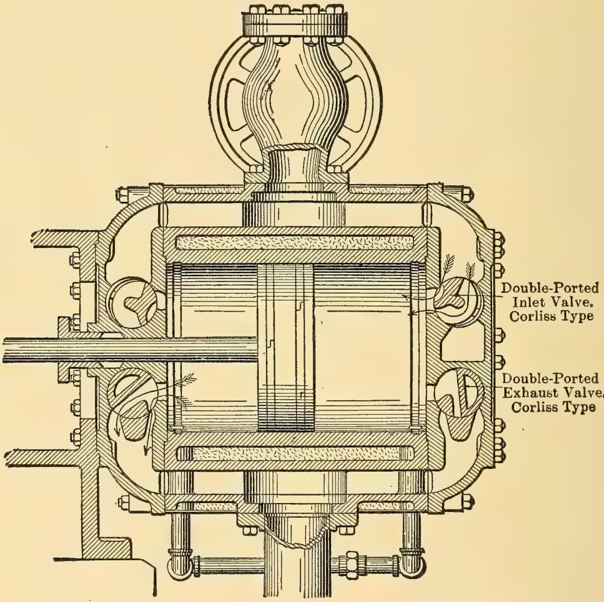


FIG. 135.—Non-detaching Corliss Valves Located in Cylinder Head.

the cylinder heads as shown in Fig. 135. In some the wrist plate and the connecting links are also retained, but in others they are eliminated. In all engines of this type the admission valves are closed positively, the closure being effected by the same linkage that opens the valves to admit steam. Quick action is obtained by the arrangement of the operating mechanisms, the centers of rotation and the

lengths of links being so chosen that the valve travel is small when the valves are closed, that it is rapid when the valves are opening and closing, and that the valves remain practically wide open during most of the time that steam is being admitted.

The advantages of small clearance and short and separate ports are attained in these arrangements and the operation of the valves is almost as perfect as that of the trip-cut-off gear. Engines fitted with these modified Corliss gears are operated at speeds considerably higher than those permissible with the older arrangement, and they may be classed with medium-speed engines.

Engines of this type are generally known commercially as **four-valve engines,** but as this name applies equally well to the ordinary trip-cut-off gear and to others which will be described later, it is best to use some other designation. The term **non-detaching Corliss engines** seems to best describe them and is apparently gaining in favor.

Non-detaching Corliss engines generally give diagrams intermediate between those obtained with the low-speed, trip-cut-off mechanism and those obtained from slide-valve engines with the simpler forms of valves, though the later designs very closely approximate the performances of the trip-cut-off Corliss engine.

92. Poppet Valves. Attention has already been called to the fact that the use of highly superheated steam is very effective in lessening or even eliminating initial condensation. Experience has shown that it is very difficult to make large valves with sliding surfaces, such as Corliss valves, work well with highly superheated steam. The large castings warp so that contact surfaces do not remain true and the lack of moisture which acts as a seal with saturated steam leads to excessive leakage. Difficulty has also been experienced with the lubrication of these sliding types of valves when using highly superheated steam.

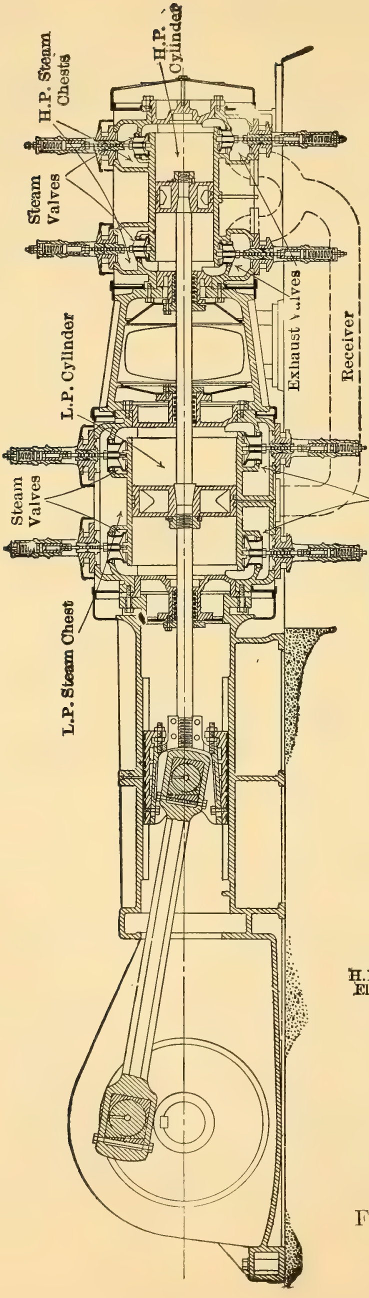
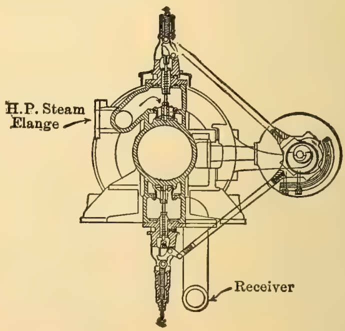


Fig. 136*a*.—Lentz Poppet Valve Engine.

An old form of valve known as the **poppet value has** therefore been adopted by some builders as a solution of the difficulties met in the use of highly superheated steam. This form of valve in four-valve arrangement, combined with designs in which short ports and symmetrical cylinder castings are used, yields very economical engines which can be used safely with a degree of superheat prohibitively high in the case of the sliding and oscillating forms of valves.



Fig, 136*b*.—Cross-section, Lentz Engine.

Sections of a modern type of poppet valve engine are shown in Figs. 136 (*a*) and 136 (*b*), and details of the admission valve and its operating mechanism are given in Fig. 137 (*a*) and (*b*). The valves are all double-seated (double-ported or double-beat), that is, they seat at both ends and are made hollow so that the steam passes both around the outside of the valve and through the valve as shown by the arrows in Fig. 137 (*b*). This results in large area for passage of steam and in quick opening and

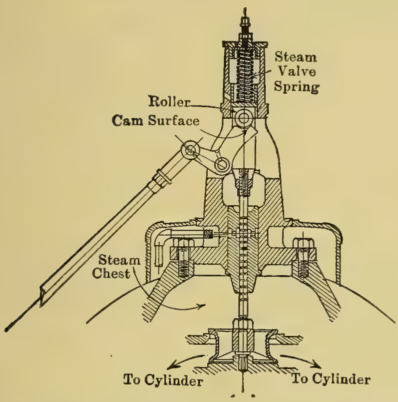


FIG. 137*a*.—Admission Valve and Operating Mechanism, Lentz Engine.

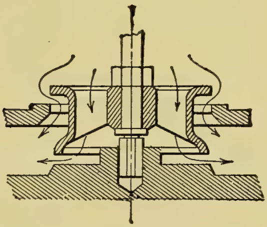


FIG. 137*b*.

closing, as in the case of gridiron valves, with small actual movement of the valve.

The valves are opened positively by eccentrics operating through cams and rollers as shown in Fig. 136 (*b*) and they are closed by springs as rapidly as the return motion of the cam permits. The eccentrics are mounted on a horizontal lay shaft which is located to one side of the engine, with its axis parallel to that of the latter, and which is driven by bevel gears from the crank shaft of the engine.

Since this valve arrangement gives short steam and exhaust poits, permits the use of small clearance, and

gives fairly rapid opening and closing of valves with little throttling when open, it gives good economy when used with saturated steam. By adding superheat the economy is still further improved. The water rate of one of these engines is shown for one load in Fig. 134 (*a*). A simple, Lentz non-condensing engine is reported to have given a consumption of 16.13 lbs. of steam per horse-power hour with 92.7° superheat, and a pressure of 133 lbs., and this figure is materially lowered by compounding, higher superheat, lower back pressure, etc.

93. The Una-flow Engine. A very interesting modification of the steam engine, known as the Una-flow Engine, has been developed recently. The purpose of the design is to permit a great ratio of expansion in one cylinder while at the same time reducing the losses caused by initial condensation when such ratios of expansion are attempted in single cylinder engines of earlier types.

Experience with the earlier types has shown that as cut-off is made earlier (ratio of expansion increased) in single cylinder engines, the steam economy is improved until best results are obtained with cut-off in the neighborhood of one-quarter stroke. With earlier cut-off the loss due to initial condensation overbalances the gain which should result from greater ratio of expansion so that a net loss ensues.

The construction of the Una-flow Engine Cylinder is shown diagrammatically in Fig. 138. Steam enters the cylinder head at *A* and jackets the entire cylinder end as it flows toward the inlet valve located in the upper part of the head. The inlet valve is a double-seated poppet valve, the seats being indicated at *B* and *C*. Exhaust occurs through openings in the cylinder wall at the middle of its length, the engine piston serving as a valve to cover and uncover these openings.

In operation, steam is admitted through the inlet valve as in other engines up to the point of cut-off. After cut-off this steam expands until the piston begins to uncover the

exhaust ports, thus effecting release. After completely uncovering these ports the piston returns and covers them again, the instant of complete closure corresponding to

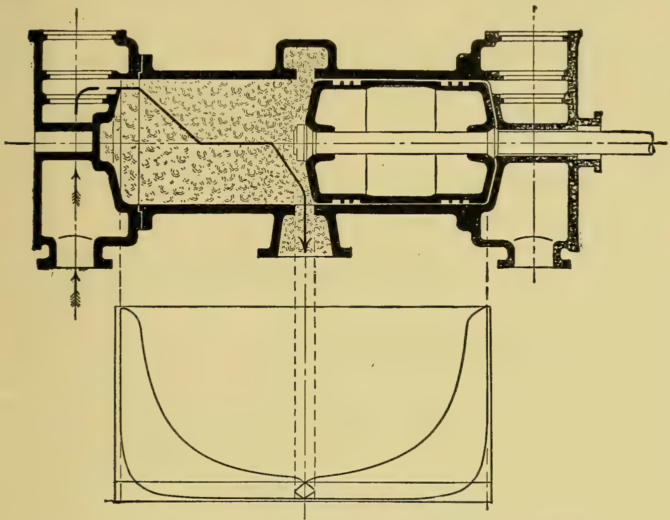


FIG. 138.—Section of Una-flow Engine Cylinder.

beginning of compression in the ordinary case. Further motion of the piston compresses the trapped steam into the clearance space, and brings conditions back to those existing when admission starts. With an ideal arrangement the pressure in the clearance space at the end of compression would be just equal to that of the high-pressure steam outside the inlet valve.

It will be observed that the piston is so long that at either end of its stroke it fills all that volume of the cylinder between the clearance and the edges of the exhaust ports. The result is that the only part of the cylinder wall which is used in common for the cycles occurring at opposite ends of the cylinder is that part containing the exhaust openings or ports. It will also be observed that this part is strictly a

low temperature zone, coming in contact only with steam at exhaust temperature and pressure.

This construction is almost equivalent to the placing of two single-acting cylinders end to end. It isolates each cylinder end in a sense, so that the thermal conditions resulting from the cycle carried through on one side of the piston are practically independent of those resulting from the cycle on the other side of the piston. In the ordinary form of double-acting engine the cylinder walls are common to both cycles for almost their entire extent with corresponding interrelation of thermal phenomena.

Another important feature is the fact that high pressure, high temperature steam enters at the cylinder head while low pressure, low temperature steam resulting from expansion flows out of the cylinder at the point most distant from the point of entry. Steam flow is thus continuously in one direction and low-pressure steam does not sweep over parts which at the next admission will be bathed with high temperature steam. As a result of this construction, of the jacketing of the cylinder head with live steam on its way to the cylinder and of the long compression giving a compression curve much nearer to the expansion curve than is commonly obtained, the temperature distribution along the length of the cylinder is much more perfectly controlled than in the ordinary case. The ends of the cylinder tend to take and retain a temperature corresponding to that of the steam supply. The middle of length of the cylinder tends to take and retain a temperature corresponding to that of the exhaust steam. The length of wall between cylinder ends and center tends to assume temperatures grading from high temperature at the ends toward low temperature at the exhaust ports.

As expansion of saturated steam occurs within a cylinder condensation occurs, work being done at the expense of the latent heat of vaporization thus released. This condensation is distributed throughout the entire mass of steam and

at the metallic walls the water has a tendency to deposit. It should be noted that the temperature at which such water is formed must correspond at any instant to the temperature of the steam, and as the temperature of the steam decreases continuously as the pressure drops during expansion, the same thing must be true of the water formed during expansion.

In the Una-flow design the cylinder head and walls near that head are kept hot by the steam jacket and if any water does deposit on them it is probably evaporated into steam

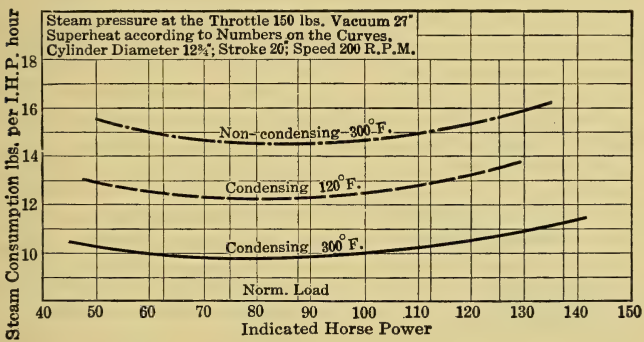


FIG. 139.—Results of Tests Made with a "Una-flow Engine."

again. The engine is assumed to operate in this way and it appears to be true that the greater part of the water resulting from expansion concentrates near the exhaust ports and is swept out during the release of steam. Such action is indicated diagrammatically in Fig. 138 by the gradation of the stippling within the cylinder.

At first sight it might appear that heat transferred from jacket to steam within the cylinder would surely result in loss but it should be noted that a large part of the steam within the cylinder which is in contact with steam-heated surfaces probably remains in the cylinder and constitutes the clearance steam so that heat transferred to it during

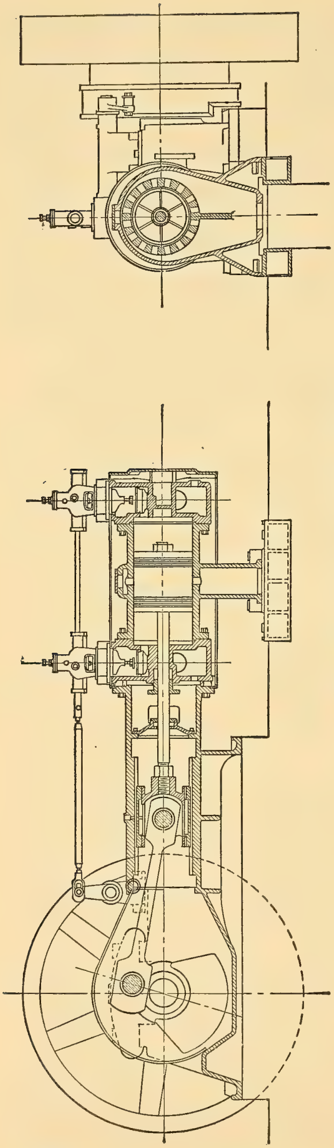


FIG. 140.—18″×20″ Una-flow Engine.

expansion does not pass out through the exhaust valve to such an extent as it would in an engine of the ordinary type.

The long compression stroke, with increasing temperature and pressure, also has a tendency to evaporate any moisture remaining on the walls. This is certainly true for that part of the metal near the cylinder head. The entering steam thus comes in contact with hot, dry walls and with hot, dry steam and initial condensation is greatly reduced, if not eliminated.

Remarkably low steam consumptions have been obtained with Una-flow engines, even when simple engines have been used to expand steam from comparatively high pressure to a high vacuum. Results obtained with one engine under different conditions are shown in Fig. 139.

The mechanical construction of an 18×20-inch single cylinder Una-flow engine is shown in Fig. 140, a section through the exhaust ports being shown to the right of the figure.

94. The Locomobile Type. In the effort to improve the ecomony of small steam plants the Germans developed a form of plant now known as the **Locomobile Type.** The name came from the fact that these plants, as originally made, were mounted on wheels and intended for portable use by agriculturists and contractors. Their economy in the use of fuel proved so great that they have since been built for stationary use in sizes running well toward 1000 horse-power per unit.

A locomobile of American construction known as the Buckeye-mobile is illustrated in Fig. 141, which shows a longitudinal section of the plant. The tandem compound engine is mounted on top of an internally fired boiler with the engine cylinders located in the flues which lead the products of combustion away from the boiler.

The steam generated in the boiler is passed through a superheater suspended in the smoke box. The flow of steam is from the rear toward the front of this superheater (counter flow) so that the hottest steam comes in contact with the hottest gas. The steam then passes through a pipe contained within the flue to the high-pressure cylinder, which is jacketed by the hot flue gases and in which the loss of heat to metal is thus minimized. From the high-pressure cylinder the steam passes to a receiver contained in the smoke box, the receiver serving as a reheater to evaporate any condensate exhausted from the first cylinder and to superheat the steam admitted to the low-pressure cylinder. From the low-pressure cylinder, the steam passes through a feed-water heater in which it raises the temperature of the boiler feed and then it passes to atmosphere or to a condenser. Boiler-feed pump and condenser pump, if used, are also integral parts of the plant, being driven directly from the main engine.

It will be observed that every precaution is taken to guard against initial condensation, and to minimize loss of heat in flue gases and in exhaust steam leaving the

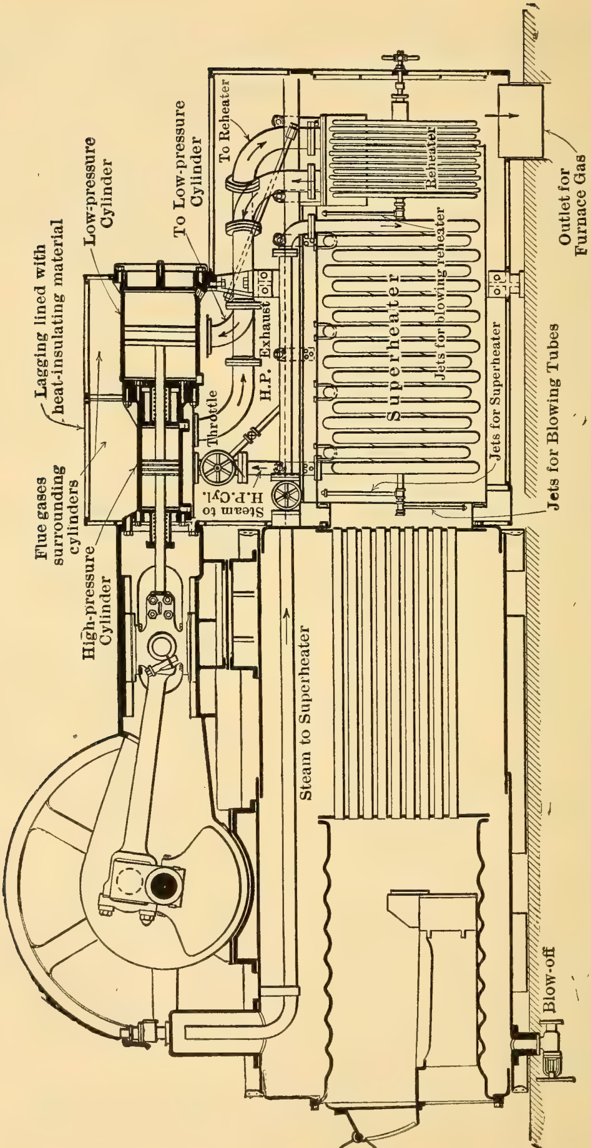

Lagging lined with heat-insulating material
Low-pressure Cylinder
To Low-pressure Cylinder
To Reheater
Reheater
Outlet for Furnace Gas
Jets for blowing reheater
Superheater
H.P. Exhaust
Throttle
Steam to H.P.Cyl.
Jets for Superheater
Jets for Blowing Tubes
Flue gases surrounding cylinders
High-pressure Cylinder
Steam to Superheater
Blow-off

FIG. 141.

plant. The high economies achieved are due to such facts alone.

Small plants of this type have given an indicated horsepower hour on a little over one pound of coal when operated condensing, whereas the best large compound reciprocating engine plants seldom do better than about 1.75 lbs. of coal per I.h.p. and often use 2 or more pounds when operated condensing.

CHAPTER XII

REGULATION

95. Kinds of Regulation. There are two distinctively different kinds of regulation referred to in connection with reciprocating steam engines, one of which may be called **fly-wheel-regulation** and the other **governor-regulation** or **governing.**

The regulating effect of the fly-wheel has already been referred to. The turning effort exerted at the crank pin by the action of steam on the piston or pistons of an engine is not constant, and the *angular velocity* of the engine shaft is therefore constantly varying during each revolution. It is the function of the fly-wheel to damp these variations so that they do not exceed the allowable maximum for any given set of operating conditions. The efficiency of the fly-wheel in this respect is measured by the coefficient of fly-wheel regulation δ_W which is defined by the equation

$$\delta_W = \frac{V_{\max} - V_{\min}}{V}, \quad . \quad . \quad . \quad . \quad . \quad (66)$$

in which

$V_{\max}$ = maximum velocity attained by a point on fly-wheel rim or other revolving part;

$V_{\min}$ = minimum velocity of the same point, and

V = mean velocity of the same point

$= \dfrac{V_{\max} + V_{\min}}{2}$ approximately.

Governor-regulation is absolutely different. Its function is to proportion the power made available to the instantaneous demand. The fly-wheel takes care of variations

occurring during the progress of one cycle, while governor regulation varies the work value of successive cycles.

96. Governor Regulation. If the effect of engine friction be neglected, the power delivered at the shaft of the engine will vary directly with the indicated horse-power. Such an assumption is accurate enough for the discussion which follows.

The indicated horse-power of a given engine is determined entirely by the value of the mean effective pressure and the number of cycles produced in a given time, since these are the only variables in the formula for indicated horse-power. The power made available by an engine might therefore be varied by varying the mean effective

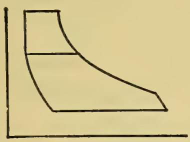


FIG. 142.—Throttling Governing.

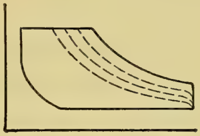


FIG. 143.—Cut-off Governing.

pressure, or by varying the number of cycles produced in a given time, or by a combination of both processes.

All of these possibilities are used. In ordinary stationary power plants the mean effective pressure is generally varied. In the case of pumping engines, working against a constant head, but required to deliver different quantities of water at different times, the number of cycles per minute is generally altered by changing the speed at which the engine operates. In locomotive and hoisting practice both the number of cycles per minute (speed) and the mean effective pressure are varied as required to meet the instantaneous demands.

These variations may be effected manually as by the driver of a locomotive, in which case the engine may be said to be **manually governed.** Or, they may be brought

about mechanically, as in the case of most stationary power-plant engines, in which case the engine may be said to be **mechanically governed**. In some instances a combination of manual and mechanical governing is used.

97. Methods of Varying Mean Effective Pressure. The mean effective pressure increases and decreases with the area of an indicator diagram of constant length, so that the mean effective pressure can be changed by any method which will change the area of the diagram. Two methods are in use and they are illustrated in Figs. 142 and 143. The first causes a variation in area by changing the value of the initial pressure. This is generally done by changing the opening of a valve in the steam line just outside of the steam chest. It is called **throttling governing,** and the valve is called a throttling or **throttle valve**. The latter name is also commonly used for the valve located near the engine, which is used to shut off the supply of steam entirely when the engine is not in operation.

The second method, illustrated in Fig. 143, is known as **cut-off governing**. The variation of cut-off determines the amount of steam admitted to the cylinder per cycle and is used to measure out the quantity required for the load which happens to exist at any instant. Cut-off governing is used on most modern stationary engines and is exclusively used in large reciprocating engine power plants.

98. Constant Speed Governing. Most engines used for such purposes as the operation of mills and the driving of electrical and centrifugal machinery are required to run at practically constant speed irrespective of the load. They are furnished with mechanical governors which so regulate the power made available that there shall never be any appreciable excess or deficiency which would respectively cause an increase or a decrease in speed.

These mechanical devices always contain some sort of tachometer which moves whenever the speed of the engine exceeds or falls below the proper value. The tachometer

is so connected to the valve gear that it decreases the power-making ability of the engine whenever the speed starts to increase and it increases the power-making ability if the speed drops.

Since the valve gear must have a different position for each load in order that it may throttle or cut off as necessary to suit that load, it follows that the tachometer which controls the position of the valve gear must also have different positions for different loads. But tachometers assume positions dependent on speed, and therefore different loads can only be obtained if the tachometer and the engine to which it is connected operate at different speeds for different loads.

Constant-speed governing is therefore an anomaly. The device which is supposed to maintain constant speed irrespective of load must be operated at different speeds, as the load varies, in order that it may maintain the valve gear in the different positions required to handle the different loads. All so-called constant-speed engines have their highest speed when carrying no load, and the speed gradually decreases to a minimum as the load increases to a maximum. The total variation is generally between 2 and 4%.

The efficiency of a governor in this respect is measured by means of the **coefficient of governor regulation**, δ_G, which is defined by the equation

$$\delta_G=\frac{n_2-n_1}{n}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

in which

n_2 = highest rotative speed attained by the engine;
n_1 = lowest rotative speed attained by the engine, and
n = mean speed
$=\frac{n_2+n_1}{2}$ approximately.

99. Governors. The mechanical devices which are used for controlling the power-making ability of an engine

as described above are known as governors. There are many varieties and only a few of the more prominent can be described.

(*a*) **The Pendulum Governor.** One of the earliest forms of governor used on steam engines is illustrated in Fig. 144. It is often called a fly-ball governor. This governor is driven by gearing, chain or belt from the engine, and the weights assume some definite position for each different speed, thus drawing the collar to different positions. The valve gear is connected to this collar and is moved correspondingly.

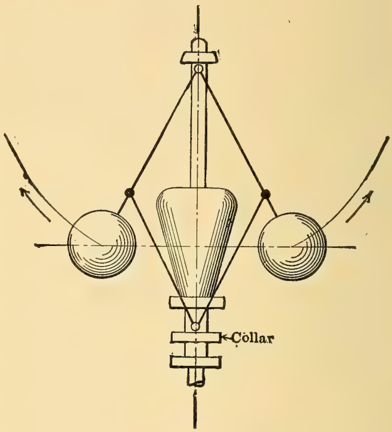


Fig. 144.

A similar governor is shown in Fig. 131, which also indicates the way in which the collar is connected to the valve gear in the Corliss type of engine. The governor rods are moved as the collar moves and they in turn alter the position of the knock-off cam, and thus vary the time at which cut-off occurs. As the speed increases due to a decrease of load, the governor weights and collar move up, and this shifts the cams so as to produce earlier cut-off and decrease power-making ability.

(*b*) **Shaft Governors.** On medium- and high-speed engines fitted with some form of slide valve it is found best to use what are known as shaft governors. They are generally carried within the fly-wheel of the engine, operate in a plane passing through the rim of the wheel at right angles to the shaft, and operate upon the eccentric in such a way as to vary the cut-off with speed (and load) changes.

One simple form of such a governor is shown in Fig. 145. The eccentric is not mounted directly upon the engine shaft, but is carried by a pin P in the fly-wheel and is slotted so that it can swing back and forth across the shaft, about P as a center. Its position at any time is determined by the position of the governor weights W, which draw the eccentric down (in the figure) as they move out.

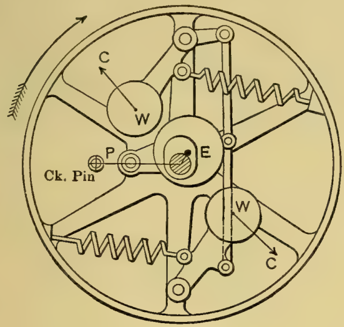


FIG. 145.

The center of the eccentric is indicated by a heavy dot in the figure, and it will be seen that this center would travel in the arc of a circle about P, as the weights moved. If the path of the eccentric center is drawn on a Bilgram diagram, it will be found that this motion is equivalent to decreasing the length of the eccentric crank and increasing the angle of advance, resulting in earlier cut-off as the weights move out with increasing speed and decreasing load. Other events will also be changed as the eccentric swings, and some of these changes are occasionally undesirable.

Numerous designs have been developed in which the eccentric is so guided as to produce various sorts of relations between the different steam and exhaust events. All can be divided into two classes, those in which the eccentric swings about a fixed center variously located, and those in which the center of the eccentric is guided to move in a straight line. All can be studied by plotting the path of the eccentric center (path of Q) on the Bilgram diagram.

The Rites Inertia Governor is a form of shaft governor so designed as to act very quickly with change of speed,

and to be very powerful, so that it can shift heavy parts. It is shown in place in the wheel in Fig. 146. With changes in speed it acts like a governor of the type just described, swinging (with increasing speed) about a fixed point P in the wheel as its center of gravity G moves outward under the action of the centrifugal effect C and against the action of the spring. This motion shifts the center of the

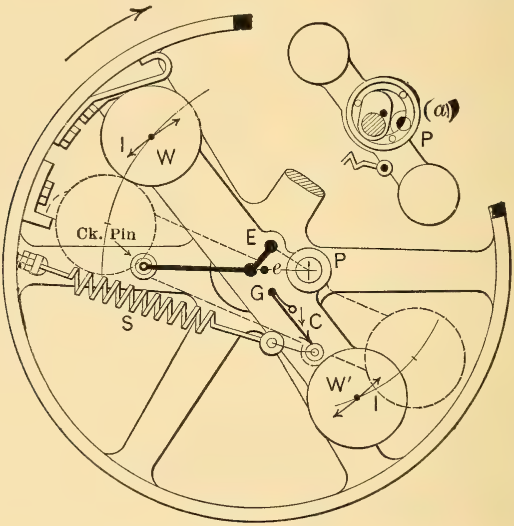


FIG. 146.

eccentric from E toward e, giving the desired variation in cut-off.

Superposed upon this action is that of inertia. Assume the wheel and governor to be rotating clockwise at a given constant speed. If the engine speed is suddenly increased, the wheel will move faster, but the governor bar will tend to continue rotating at the same speed because of its inertia. It will thus lag behind the wheel, rotating about P and bringing about an earlier cut-off. The position thus assumed will later be maintained by centrifugal effect if the new speed

is maintained. The particular advantage resulting from using inertia in this way is speed of action. In many forms of governor the inertia of the moving parts actually resists the efforts of the governor to assume the new position required by changed load and speed, whereas in this form the inertia of the governor parts is used to increase the speed with which they move to the new position.

CHAPTER XIII

THE STEAM TURBINE

100. The Impulse Turbine. One of the oldest of modern water wheels is the tangential or impulse wheel shown diagrammatically in Fig. 147. Water flowing from a reservoir above the wheel passes through a nozzle and the

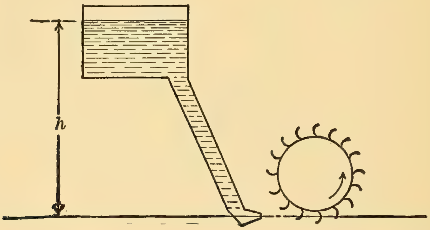


FIG. 147.—Tangential or Impulse Wheel.

jet, moving at high velocity, strikes buckets on the rim of the wheel and causes the latter to revolve. Theoretically the velocity of the water in the jet would be

$$v = \sqrt{2gh} \text{ feet per second}, \quad . \quad . \quad . \quad . \quad (68)$$

in which

g = gravitational constant, 32.2, and
h = head in feet as shown in the figure.

The kinetic energy possessed by the moving water would be

$$K = \frac{w}{g} \cdot \frac{v^2}{2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

in which w represents pounds of water discharged per second and g and v have the same meanings as above.

If the buckets of the wheel could reduce the velocity of the water to zero they would absorb all of this kinetic energy and (assuming no losses within the buckets and the bearings of the wheel) would make all of it available at the shaft for the doing of useful work.

Any fluid moving at velocity v and striking buckets in the form of a jet would possess kinetic energy in quantity given by Eq. (69) and would drive the wheel in the same way. Steam might therefore be used instead of water with exactly the same results, and steam is so used in what are known as **impulse steam turbines.**

Experience shows that steam will flow at high velocity from any opening made in the steam space of a boiler or from any open-ended pipe connected to such a boiler. This is commonly said to be due to the high pressure within the boiler, the spectator picturing the process as the driving out of part of the steam by the high-pressure steam within the boiler, just as though the part leaving were a solid piston and were driven out as is the piston of an engine during admission, as shown in Fig. 148.

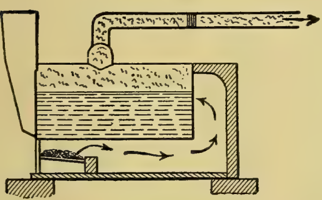


FIG. 148.

An hydraulic analogy is given in Fig. 149. The vessel shown is supposed to be fitted with a piston, and it is assumed to be possible to exert any desired pressure upon the piston. Any such pressure exerted is the exact equivalent of some given head of water and the resultant jet velocity would be given by Eq. (68) by substituting for h the head in feet equivalent to the pressure exerted upon the piston.

FIG. 149.

When an "elastic" fluid such as steam is being con-

sidered it is, however, necessary to take account of other factors. The steam within the boiler exists at a high pressure; after issuing it exists in the atmosphere at a lower pressure. But low-pressure steam contains less heat than does steam at high pressure, and this difference must exist in some form, as it is energy and could not possibly have been destroyed during the flow.

Experiment shows that steam after flowing into the atmosphere from a boiler in this way has exactly the same characteristics as though it had expanded adiabatically behind a piston through the same temperature range, excepting for the fact that it has a very high velocity, which it would not possess if expanded behind a piston. Experiment further shows that, if small losses be neglected, the kinetic energy possessed by a jet of steam is exactly equal to the energy which would be turned into work if that steam acted on a piston as in an ordinary engine.

A complete picture of the process of flow can then be made by assuming the steam flowing out in the form of a piston driven by high-pressure steam, as before, and adding to this the idea that this piston expands adiabatically as it travels from the region of high to that of low pressure. This expansion liberates heat contained within the piston or plug of steam and this heat is used in imparting additional velocity to the moving steam which is giving up this heat.

The result of using such a jet upon a theoretically perfect tangential or impulse wheel would be to rob the jet of all this energy. But the energy possessed per pound of steam in the jet is just the same as that shown under the upper lines of a complete expansion cycle using one pound of steam. The area under the upper horizontal line of the PV-diagram of the cycle as shown in Fig. 21 may be assumed to represent the work done upon one pound of steam (flowing out) by another pound which is being evaporated and pushing out the first in order to make room for itself. The

area under the expansion curve in the PV-diagram represents the energy converted into velocity energy by the adiabatic expansion of the flowing steam. The lower horizontal line represents the negative work during condensation to water at the lowest pressure and temperature, and the left-hand line represents the pumping of this water back into the boiler and the raising of its temperature to the value

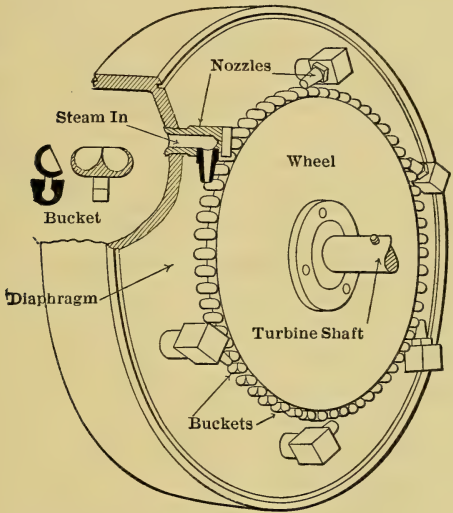


FIG. 150.—Early Form of Impulse Turbine.

maintained within the boiler. The *complete expansion cycle is therefore the cycle upon which the impulse steam turbine operates* and, as a matter of fact, it is the *theoretical cycle of all steam turbines.*

The ideal impulse turbine would therefore be acted upon by a jet which possessed available kinetic energy represented by the area of the complete expansion cycle. If the buckets could entirely remove this energy, that is, could reduce the velocity of the jet to zero, the same amount

of energy could theoretically be made available at the shaft of the turbine.

An example of a simple form of impulse steam turbine is given in Fig. 150, in which the essential parts of an early form of Kerr turbine are shown. The wheel, the diaphragm and nozzles are all inclosed within a casing. The space on one side of the diaphragm is connected to the steam pipe and that on the other is in communication with the space into which the exhaust steam is to be exhausted.

Another form of impulse turbine is shown in Fig. 158. It will be described later.

101. Theoretical Cycle of Steam Turbine. It was shown in the preceding section that the steam turbine operates on the complete expansion cycle. If a turbine could remove from the steam passing through it and convert into mechanical form all the energy which it is theoretically possible to convert it would therefore make available mechanical energy represented by the area of the PV-diagram of the complete expansion cycle. The area of the corresponding $T\phi$-diagram would show the same quantity measured in thermal units. The theory of the steam turbine can therefore be studied by means of these two diagrams.

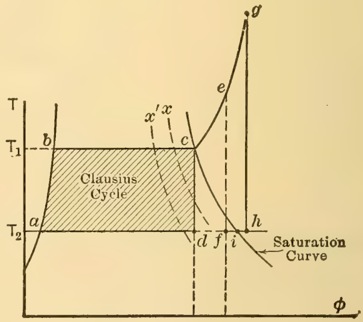


Fig. 151.

In Fig. 151 is shown the $T\phi$-diagram of the complete expansion cycle for several different conditions. The figure *abcd* represents conditions when the steam is dry and saturated at the beginning of the adiabatic expansion *cd*. Constant quality lines are designated by x and x'. It is obvious that by the time the steam has expanded down to the pressure at d it will have a quality

less than unity. If, therefore, it be in the form of a jet issuing from a nozzle and having a high velocity by virtue of its adiabatic expansion, the jet will really be a mixture of steam and water.

If the steam be superheated at constant pressure as shown by *ce* before passing through the nozzle, it is evident from the figure that the jet issuing from the nozzle will contain less water than in the preceding case, because the condition of the material in the jet after adiabatic expansion will be as shown at *f* instead of as shown at *d*. The cycle in such a case would also be larger by an amount indicated by the area *cefd*, representing just that much more heat converted into mechanical energy per pound of steam or other unit for which the diagram happened to be drawn.

If superheating had been carried to the point indicated by *g* before expansion, the jet would obviously issue from a nozzle in the form of superheated steam as shown by the point *h* in the figure. In that case the cycle would be *abcgha*, and superheat would have to be removed from the low-pressure steam to bring it to the conditions indicated at *i* before condensation could begin.

If desired, the *PV*-diagrams for such cycles can be drawn very easily. The line *bc*, or *be* or *bg* is a horizontal line in the *PV*-diagram. The line *ha* is similarly horizontal and the line *ab* is vertical. The adiabatic expansion is represented by a curved line in the *PV*-diagrams, but can be drawn easily because the necessary data are obtainable from the Tϕ-diagram, in which this expansion is represented by a straight line.

ILLUSTRATIVE PROBLEM

Draw the *PV*-diagram for a steam turbine receiving one pound of steam at a pressure of 200 lbs. absolute. with a temperature of 500° F. and exhausting against a pressure of 0.5 lbs. absolute.

First, locate on a $T\phi$-chart for steam the point representing the condition of steam at 200 lbs. pressure with a temperature

of 500° F., and draw a vertical line extending downward until it cuts the horizontal temperature line corresponding to

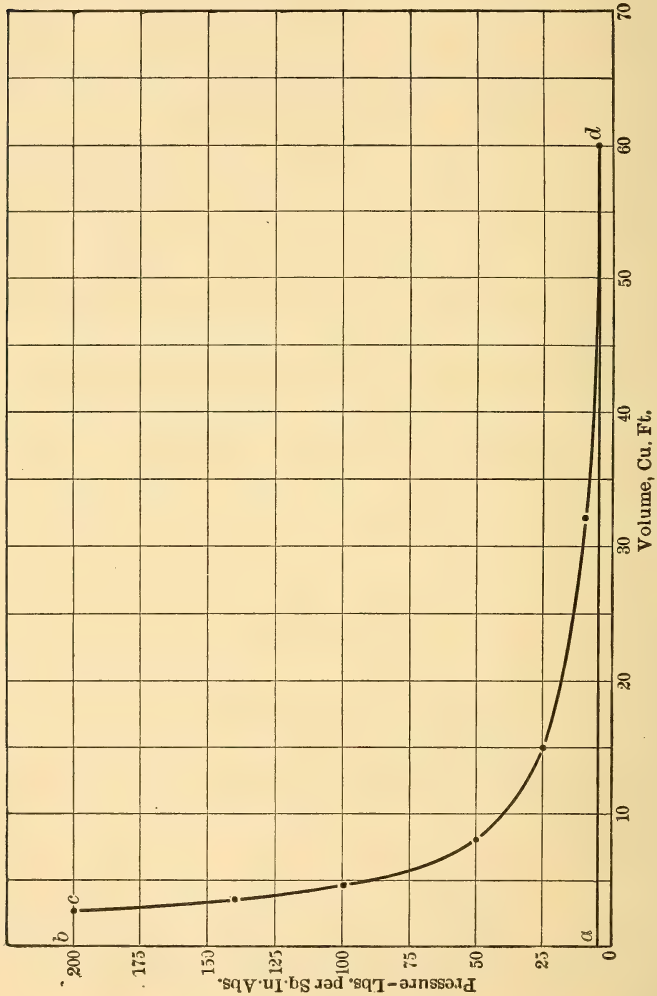


FIG. 152.

0.5 lb. pressure. This is practically at 540° F. absolute, or about 80° F.

Second, take from the steam table the volumes of one pound of steam at, say, 200 lbs., 140 lbs., and 100 lbs. absolute pressure when superheated to the values shown by this vertical line. These will be about 2.75 cu.ft., 3.58 cu.ft., and 4.67 cu.ft., respectively. Plot these volumes with corresponding pressures on a PV-chart as shown in Fig. 152.

Third, take from the $T\phi$-chart the pressures at which the vertical line intersects different volume lines in the wet steam region and plot volumes against pressures on the PV-chart.

Fourth, draw a smooth curve, as shown, through all points so determined.

Fifth, draw horizontal top and bottom lines and a vertical line at the left of the diagram. This vertical line should be to the right of the pressure axis by an amount representing the volume of one pound of water, but the volume is so small that it cannot be plotted to any ordinary scale.

102. Nozzle Design. It was stated in preceding sections that the energy which would be converted into work by the introduction and adiabatic expansion of steam behind a piston is converted into kinetic energy when steam flows out of an orifice or nozzle and that an ideal impulse turbine could absorb all this kinetic energy from the jet, bringing it to rest and making the energy available in the form of useful power at its shaft. It is, therefore, of interest to determine the velocity which a jet will acquire under different conditions.

This could be done by evaluating the area of a diagram, such as that of Fig. 152, and then putting this value in place of K in Eq. (69) and solving for v, but it can be done much more accurately and expeditiously in other ways. The heat energy which can be converted into kinetic energy of the moving jet and which can later be converted into useful work by the turbine wheel is represented by the area enclosed within the lines of the complete expansion cycle when drawn on the $T\phi$-diagrams. That is the area $abcd$ in Fig. 153, for instance, for the case of wet steam at the beginning of expansion. But this area is equal to that representing the heat supplied minus

that representing the heat rejected, that is, $Q_1 - Q_2$, so that

$$K(\text{in B.t.u.}) = Q_1 - Q_2. \quad . \quad . \quad . \quad . \quad . \quad (70)$$

The values of Q_1 and Q_2 can be found very readily by plotting the points c and d upon a $T\phi$-chart for steam and observing the constant heat lines upon which they fall,

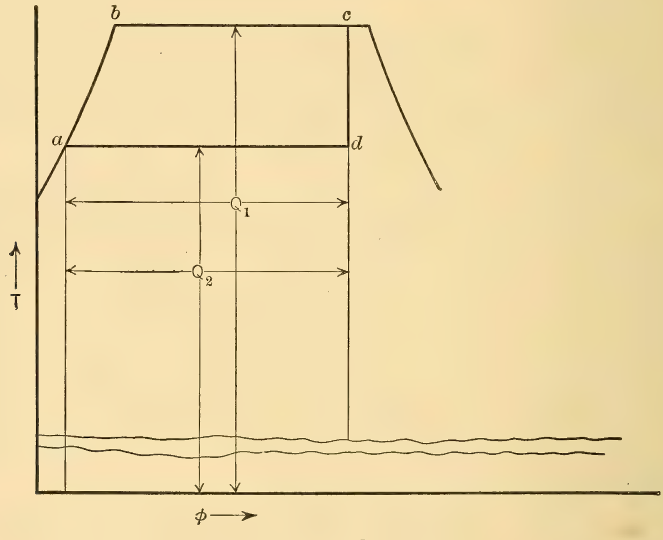
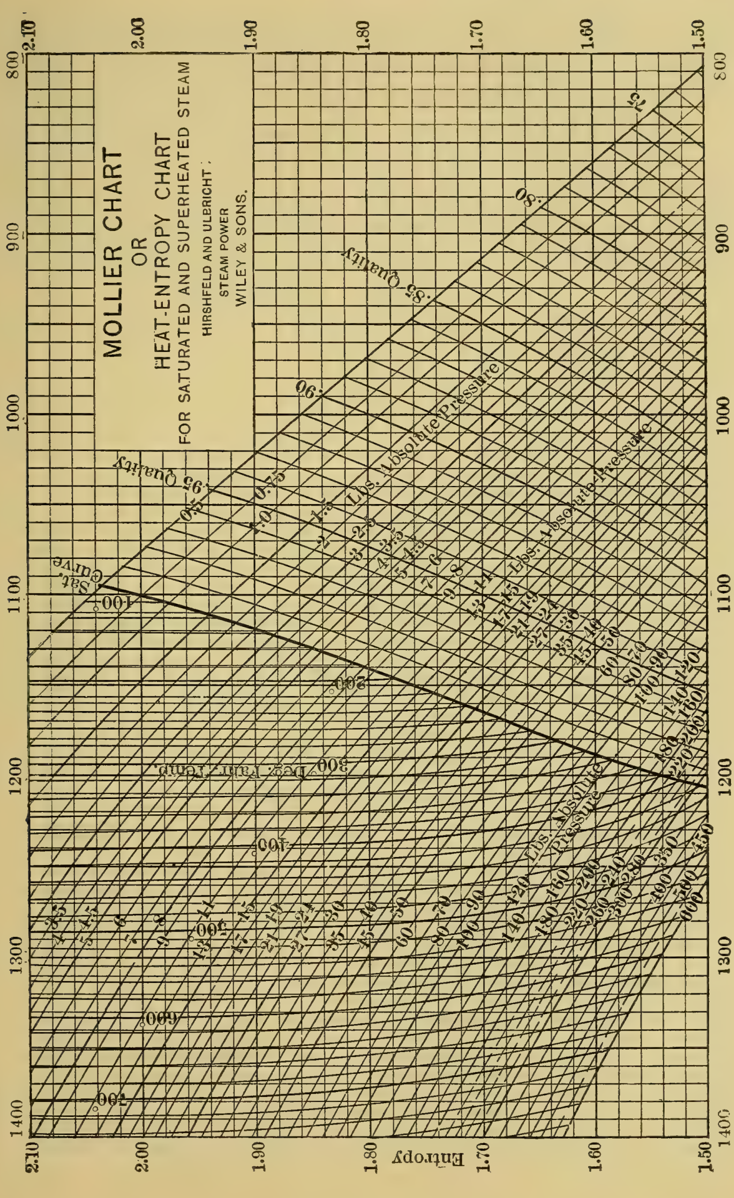


FIG. 153.

or they can be obtained even more conveniently from what is known as a Mollier Chart for steam. In this chart, entropy above 32° F. is plotted against heat above 32° F. as shown in Fig. 154. An adiabatic expansion on this chart is shown by a horizontal line, since this shows a constant entropy change just as a vertical line on the $T\phi$ chart shows a constant entropy change.

If a point is found in this chart giving conditions corresponding to those at point c in Fig. 153, the value of Q_1



B.T.U. Above 32° per Pound of Steam
Fig. 154.

can be read directly under that point on the horizontal axis. A horizontal line drawn from that point to the terminal-pressure line will give the point corresponding to d of Fig. 154 and the value of Q_2 can be read on the horizontal axis immediately below that point. The difference between the two readings gives the value of the kinetic energy K or of the mechanical energy which an ideal turbine could make available, but the expression will be in British thermal units and not in foot-pounds.

This value of the kinetic energy, i.e., $K=Q_1-Q_2$, may then be placed in Eq. (69), giving,

$$K=778(Q_1-Q_2)=\frac{v^2}{2g}\text{ ft.-lbs.,} \quad . \quad . \quad . \quad (71)$$

since Q_1 and Q_2 refer to one pound of steam, or

$$K=778w(Q-Q_2)=\frac{wv^2}{2g}\text{ ft.-lbs.,} \quad . \quad . \quad . \quad (72)$$

when w represents the number of pounds of steam flowing per second.

Solving cither Eq. (71) or Eq. (72) for v gives,

$$\begin{aligned} v&=\sqrt{778\times 2g(Q_1-Q_2)} \\ &=\sqrt{778\times 64.4(Q_1-Q_2)} \\ &=224\sqrt{Q_1-Q_2}\text{ feet per second.} \quad . \quad . \quad (73) \end{aligned}$$

The design of a nozzle consists simply in choosing such sections that the desired amount of steam may flow through it with the desired pressure drop, as the velocity obviously is determined by that pressure drop. This is very conveniently done by working in terms of one pound of steam, since all formulas and charts are generally given on that basis, and then multiplying the cross-sectional areas found by the number of pounds of steam required.

Assume for instance that it is desired to design a nozzle

to pass one pound of steam per second with an initial pressure of 100 lbs. per square inch abs., and a terminal pressure of 60 lbs., the steam being initially dry and saturated.

The Mollier chart shows that Q_1 is equal to about 1187 B.t.u. per pound of steam, while Q_2 is equal to about 1147 B.t.u. The velocity with which a jet would issue from a theoretically perfect nozzle under these conditions may then be found by using Eq. (73). This gives

$$v = 224\sqrt{1187-1147}$$
$$= 1416 \text{ feet per second.}$$

The shape of the entrance end of the nozzle is generally made such that the steam will enter it without great disturbance and the shape beyond that point is determined by methods which will be explained below. The cross-section of the discharge end must be such as to pass the required quantity at the velocity found above to be equal to 1416 feet per second. This is easily done by determining the volume of steam discharged.

Drawing the adiabatic expansion on the Tϕ-chart will give the quality at the end of the expansion; or, the quality can be determined by finding what quality a pound of steam at 60 lbs. pressure must have to give it a heat content of 1147 as found above. With the quality known the terminal volume per pound can be found by multiplying the quality by the specific volume at terminal conditions. Thus for the case under discussion the quality will be about 96.7% and as the specific volume at 60 lbs. is 7.17 cubic feet, the volume to be passed per second, per pound of steam is $0.967 \times 7.17 = 6.94$ cu.ft approximately. If the velocity is 1416 feet per second the area per pound of steam must be $6.94 \div 1416 = 0.0049$ sq.ft.

The exact shape of the nozzle is determined by deciding upon the way in which pressure, or velocity, or volume

shall change as the steam passes through it. Suppose, for instance, that a nozzle is to be constructed of the length shown by *ab* in Fig. 155, and that the pressure is to vary along its length as shown. Assume also that the nozzle is to pass 10 lbs. of steam per second. Taking initial pressure as 100 lbs. and terminal as 60 lbs., the conditions

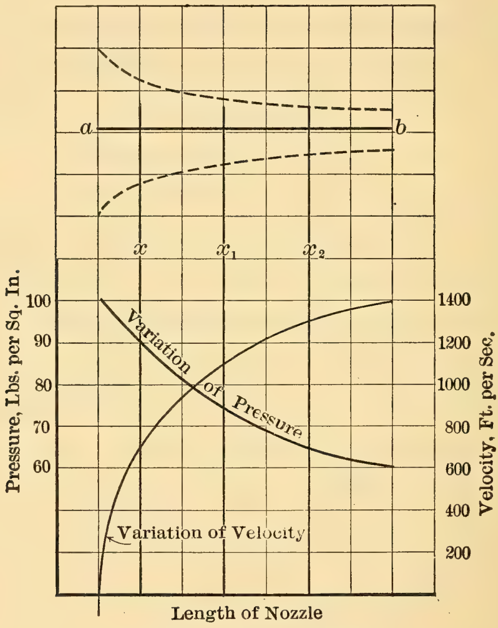


FIG. 155.—Nozzle Design.

will be the same as in the problem above. The discharge area will have to be 10×0.0049 sq.ft. or 0.049 sq.ft.

The area at the plane x_2 must be that required to pass the steam when it has the velocity resulting from expansion from 100 down to 64 lbs., just as though the nozzle ended at that point. This can be found just as the terminal area was found above. Similarly the sections at x_1 and x can be found by figuring velocity and area for expansions to

74 and 90 lbs., respectively. If the various areas required are determined in this way, the nozzle will have a longitudinal section about as shown by the dotted lines in the figure and the variation of velocity will be about as shown by the curve.

If the shape of a nozzle is determined in the same way for a case in which the terminal pressure is less than about 0.58 of the initial pressure, the nozzle will be found to have a very different shape. This is shown in Fig. 156. The nozzle is known as an expanding nozzle and the smallest section is known as the neck. The pressure P_n in the neck is always equal to about 0.58 P_1 and the velocity in the neck is always equal to just over 1400 feet per second. It is therefore the section at the neck which determines the quantity of steam which a nozzle will discharge if expanding to a pressure equal to or lower than 0.58 P_1.

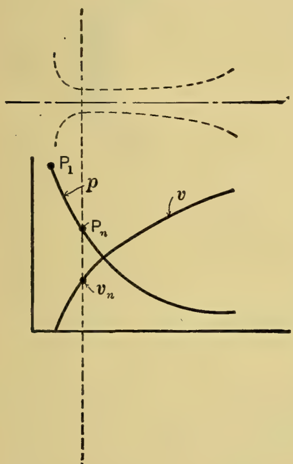


FIG. 156.—Expanding Nozzle.

103. Action of Steam on Impulse Blades. It has been stated that the steam acting in an impulse type of turbine delivers energy to the wheel of the turbine by giving up its kinetic energy. In an ideal turbine the steam jet would be brought to rest and would thus give up all of its kinetic energy.

In real turbines it is impossible to bring the jet to rest, as practical design problems prevent it. There is therefore always a loss in real machines because of the *residual* or *terminal* velocity of the steam as it leaves the wheel. Thus let the black section in Fig. 157 represent the section of a bucket or blade sticking out radially from the rim of a

wheel, the wheel revolving about the axis indicated by the dot dash line but located behind the plane of the paper, see Fig. 158. If minimum loss by eddying is to be experienced at the point at which the steam jet enters the blade, the jet must enter the blade along a tangent to the curve of the inside at the entrance edge. This direction is shown by the line marked v_r in the figure.

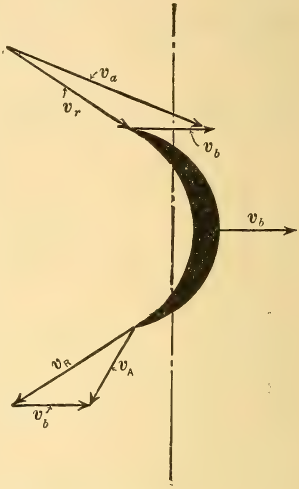


FIG. 157.

Were the bucket stationary, the steam jet would move as shown by v_r, but as the bucket moves ahead, and, so to speak, runs away from the jet, the steam must really travel in a direction such as that indicated by v_a in order to strike the bucket in the direction indicated by v_r. The conditions governing the flow of steam into a bucket are the same as those governing the speed with which and direction in which an individual runs toward and jumps upon a moving vehicle. He will experience least shock when he is moving ahead at the same rate as the vehicle at the instant when he gets on board. His motion must therefore be made up of two, one toward the vehicle and the other in the direction of the vehicle's travel.

In the case of steam flowing onto a blade as shown in Fig. 157, the various velocities are so related that when drawn to scale the real or absolute velocity of the steam, v_a, and the real or absolute velocity of the blade, v_b, form two sides of a triangle of which the closing side represents v_r, the velocity of the steam relative to the bucket. The value and direction of v_r is obviously obtained from v_a by geometrically subtracting the velocity of the bucket.

After entrance, the steam flows around the inner curve of the blade and is finally discharged with the same relative velocity as that with which it entered, and at an angle set by the tangent to the inner curvature of the discharge edge of the blade as shown by v_R. But, since the steam has been moving ahead with the same velocity as the bucket during the entire time that it was in contact with the bucket, it is also moving ahead with a velocity v_b when it leaves the wheel. Its real or absolute velocity is then v_A, which is found by combining v_R and v_b as shown in the figure.

The kinetic energy possessed by the jet when entering the blade is equal to $\frac{wv_a^2}{2g}$ ft.-lbs., and that which it possesses when leaving is $\frac{wv_A^2}{2g}$. Obviously, the energy removed while passing over the blade is $\frac{wv_a^2}{2g}-\frac{wv_A^2}{2g}$. If the blade were theoretically perfect, it would be so constructed that v_A^2 would be zero and all of the kinetic energy would then be removed. This is practically impossible in a real mechanism, and there is always a loss due to the residual velocity v_A. The best that can be done is to so choose the angle of jet and blade, and the velocity of blade with respect to the steam that the actual numerical value of v_A is made as small as possible.

Designs usually work out in such a way that this occurs when the blade velocity is equal to about 0.47 of the absolute velocity of the steam jet.

104. De Laval Impulse Turbine. The expanding nozzle already described was first used by De Laval in an impulse type of turbine. The essential elements of this device are shown in Fig. 158. The nozzles are arranged at such an angle to the plane of the wheel that the steam jets strike radially arranged blades at the proper angle to enter without much loss. The blades deflect the jets as shown and

absorb the greater part of their kinetic energy, so that the steam flows away from the wheel with low absolute velocity.

As many nozzles are used as are required to make avail-

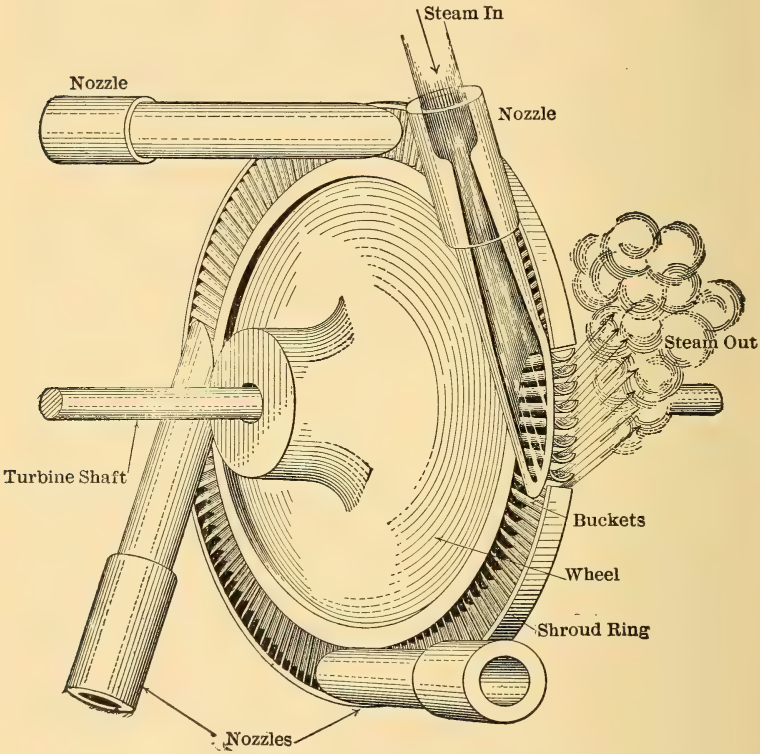


FIG. 158.—Single Stage, De Laval Impulse Turbine.

able the amount of energy desired at full load, and provision is made for shutting off one or more nozzles by hand when conditions do not warrant the use of all. Governing for ordinary variations of load is effected by throttling the steam flowing to the nozzles in use, thus altering the initial pressure as necessary.

A section through the wheel and casing of such a turbine "direct connected" to a centrifugal pump is given in Fig. 159. The steam flows into the live steam space through a throttle valve controlled by the governor; the valve and connections are not shown in the illustration. From the live steam space the steam flows through nozzles not shown, and into the exhaust steam space, thus acquiring a high velocity. The buckets of the wheel are located just in front of the discharge ends of the nozzles and the steam moving at high velocity must pass through them before moving on toward the exhaust outlet.

105. Gearing and Staging. It has been stated that the most efficient operation with ordinary designs is obtained when the blade speed is equal to about 0.47 of the absolute steam velocity or, roughly, half the velocity of the impinging jet. To get high economy in the use of steam, large pressure drops are used and very high jet velocities result. When the buckets of a turbine are operated at peripheral speeds equal to half these jet velocities one of two difficulties is often met. The stresses induced in the wheel structure by centrifugal effects become so high as to offer serious difficulties in design, or the rotative speed of the unit becomes too high for direct connection to the machine which is to be driven.

One method of partly overcoming the latter difficulty is to operate the turbine at or near the theoretically desirable speed and transmit the power to the driven machine through gears which decrease the rotative speed to the necessary extent. This method was used with all of the early De Laval turbines which were of comparatively small capacity. It is now being successfully applied to marine propulsion and other purposes for which large units are used. It is only a partial remedy in the case of large units, however, as the gears necessary for the desired reduction and the size of the turbine wheels would both become excessive.

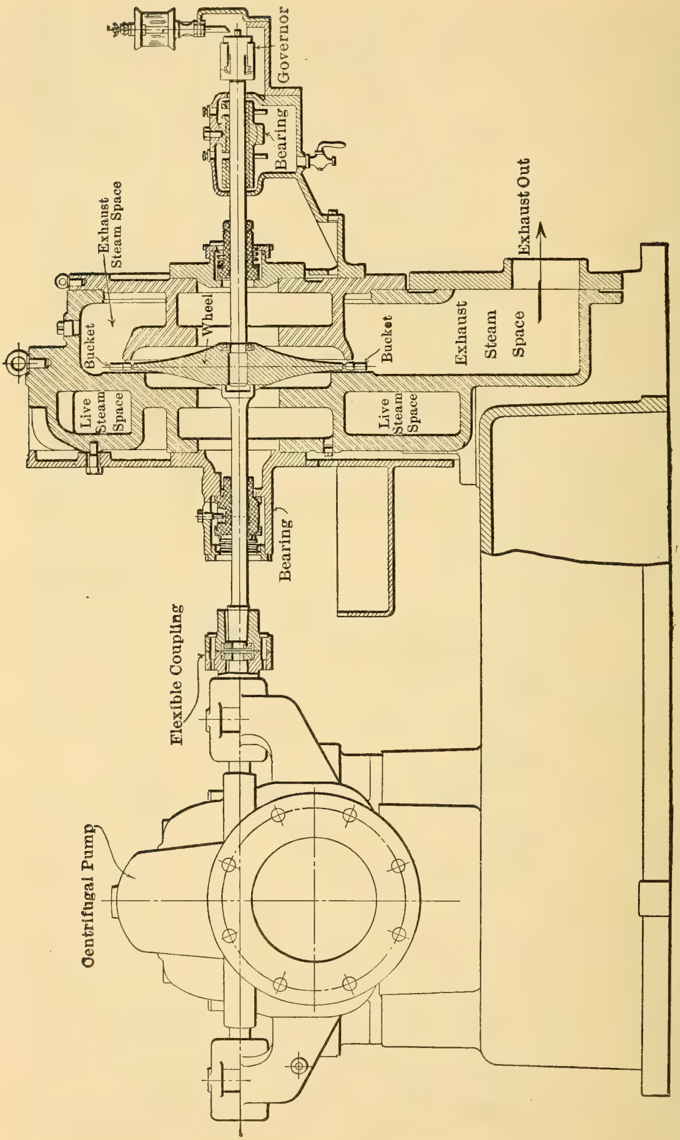


FIG. 159.—Section of De Laval Single Stage Impulse Turbine.

Another and very common method is known as *compounding* or *staging*. This may be of two varieties. The pressure drop in each stage may be limited to that which will give a reasonable velocity and a number of such stages may be put together in series on one shaft. This would give one set of nozzles and a wheel for each stage, the steam discharged from one wheel with very low velocity expanding to a lower pressure through the nozzles of the next stage and impinging upon the wheel of that stage with the resultant high velocity. Such an arrangement is known as pressure staging or pressure compounding, and is extensively used in large turbines.

The pressure staging method is illustrated in Fig. 160 as applied to the De Laval type of impulse turbine. The combined increase in diameter of wheels and increase in length of blades gives the necessary increase in area to pass the larger volumes of steam as the drop of pressure continues from stage to stage.

Instead of staging on a pressure basis, staging on a velocity basis may be used. In such a case the drop in pressure through one set of nozzles is great and the resultant velocity high. The steam moving at this high velocity is then directed upon the buckets moving at such peripheral velocity that they absorb only part of the kinetic energy of the steam, discharging it with a lower absolute velocity than that with which it entered, but one which is too high to be thrown away. The steam then passes through a set of stationary vanes which direct it upon the blades of a second wheel, in passing through which it gives up still more of its kinetic energy with a corresponding further decrease of velocity. If the velocity still possessed by the steam warrants it, a second set of stationary guide vanes and a third set of moving buckets can be supplied for further reducing it and by carrying this velocity staging through a sufficiently great number of stages any initial velocity

could be absorbed theoretically without the use of wheels with high peripheral speeds. Practically, losses due to

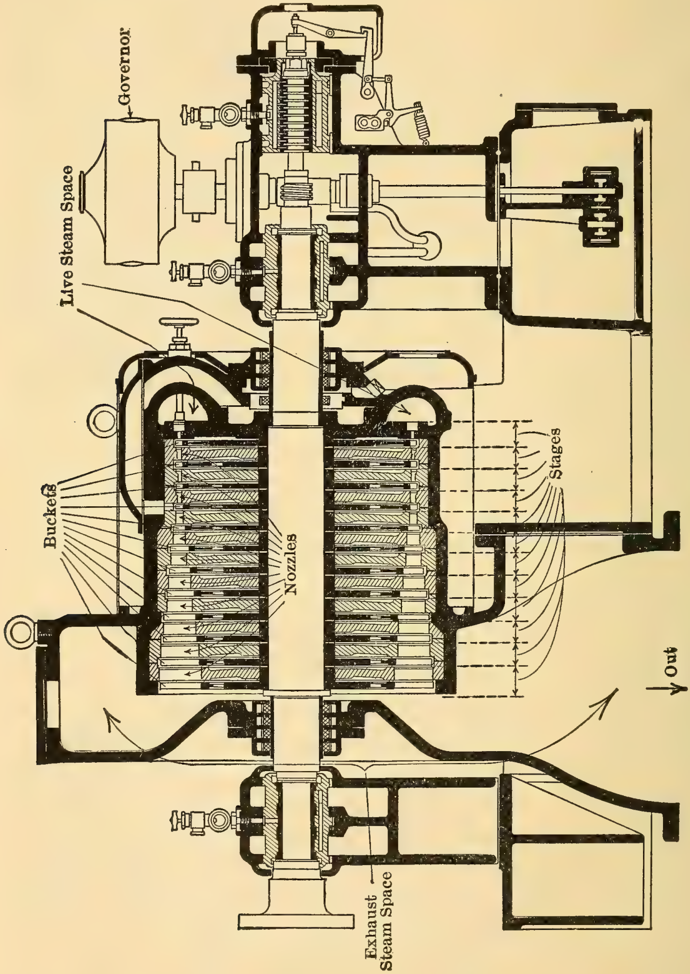


Fig. 160.—Multistage De Laval Steam Turbine.

friction, eddying and other sources limit the number of velocity stages to two or three.

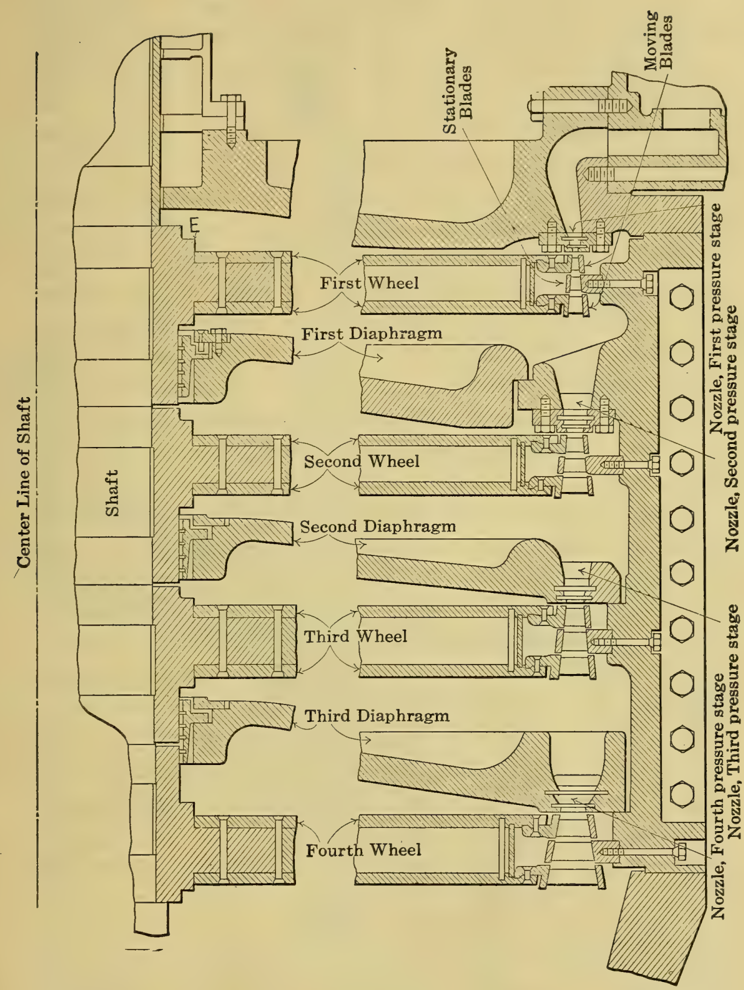


FIG. 161.—Early Form of Curtis Turbine.

Velocity staging is combined with pressure staging in the Curtis type of turbine. A section through part of an early design of vertical turbine of this type as built by the General Electric Company is shown in Fig. 161. The turbine illustrated had four pressure stages and each pressure stage had two velocity stages.

Many varieties of impulse turbines have been developed and all of the larger ones employ several wheels and sets of nozzles and diaphragms to obtain the necessary staging. The same result has been obtained in some of the smaller models by discharging the steam from nozzles on to a set of buckets which are able to absorb only a fraction of the kinetic energy, catching it at discharge and returning it for another passage through the buckets, and so on until the greatest practical fraction of the kinetic energy has been absorbed.

A vertical section through a large, horizontal turbine of the impulse type is given in Fig. 162. Units of this sort are built with different numbers of stages depending upon both the total pressure drop for which they are designed and the thermal efficiency which is desired. It is obvious that the number of stages required to give a certain thermal efficiency will increase with the total pressure drop (that is, the difference between steam pressures at entrance and exit respectively) if the peripheral speed permitted is to remain the same. Conversely, if other things remain equal, increasing the number of stages increases the thermal efficiency up to the point where increasing losses overtake further possible gains.

Commercially, the number of stages used in any given case is determined as a sort of compromise between first cost of unit, operating reliability and money value of thermal efficiency. Twenty-two stages are about the upper limit at the present time and the great majority are built with a smaller number.

The impulse type is built in all sizes between a unit

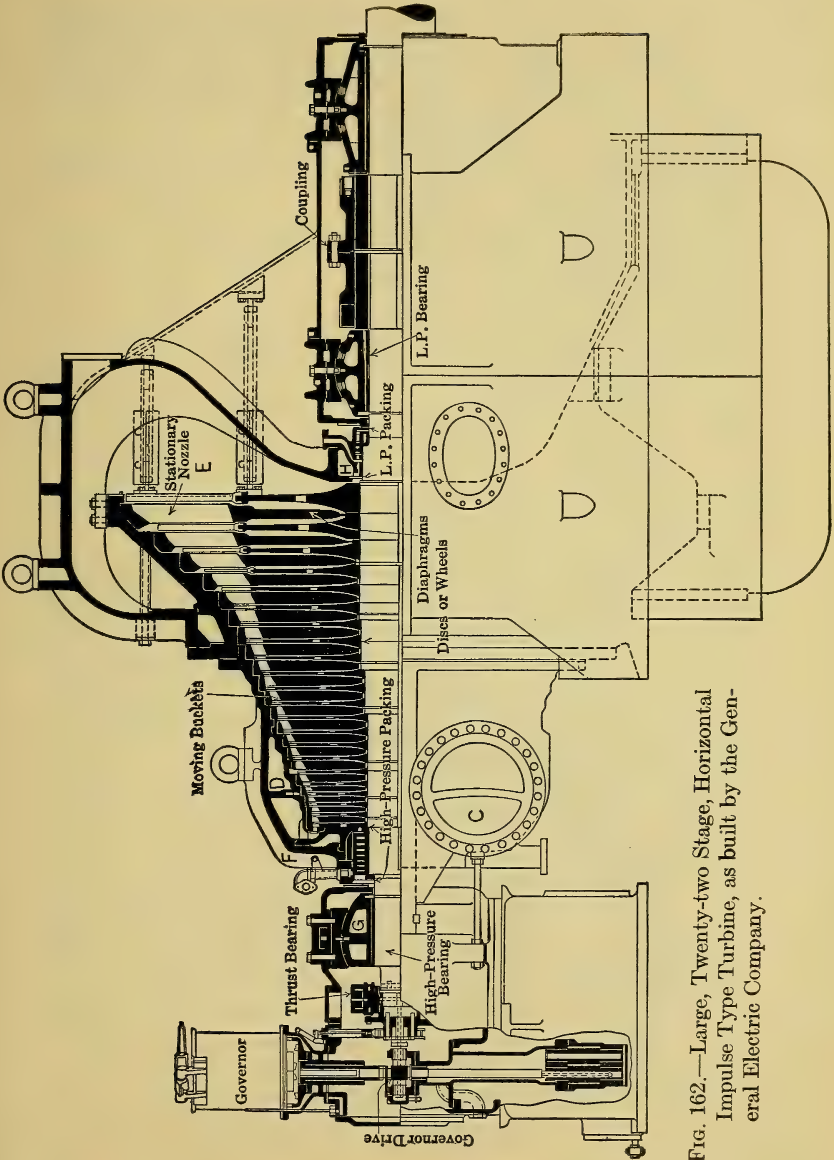


FIG. 162.—Large, Twenty-two Stage, Horizontal Impulse Type Turbine, as built by the General Electric Company.

capable of developing a few horse-power and a unit capable of developing about 60,000 horse-power.

106. The Reaction Type. If high-pressure steam or other fluid be forced into a device arranged as shown in Fig. 163 and free to revolve about a vertical axis, the jets blowing out of the nozzles will cause the mechanism to revolve in the direction indicated by the arrow. This rotation is said to be due to the *reaction* of the jets, and the mechanism therefore constitutes a simple form of **reaction turbine.** By increasing the number of nozzles any amount of steam could be discharged and therefore any amount of work could be obtained.

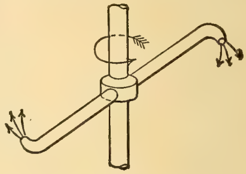


Fig. 163.
Elementary Reaction Turbine.

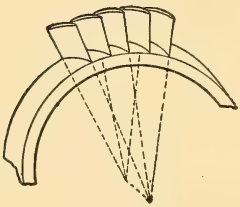


Fig. 164.

This multiplication of nozzles can, however, be more conveniently accomplished by fastening radial vanes to the periphery of a wheel as shown in Fig. 164, the space between any two vanes constituting a nozzle through which the steam can discharge. By mounting such a wheel within a casing as shown in Fig. 165 it forms **a simple reaction turbine.** One of the characteristic differences between the impulse and the reaction types lies in the distribution of pressures. In the impulse type the nozzles are fastened into a stationary part of the turbine and the drop of pressure occurs entirely within the nozzles. The wheels are therefore immersed in a space in which a uniform lower pressure exists. In the reaction type, on the other hand, the nozzles are carried on the wheel and

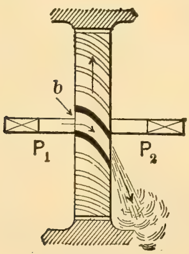


Fig. 165.

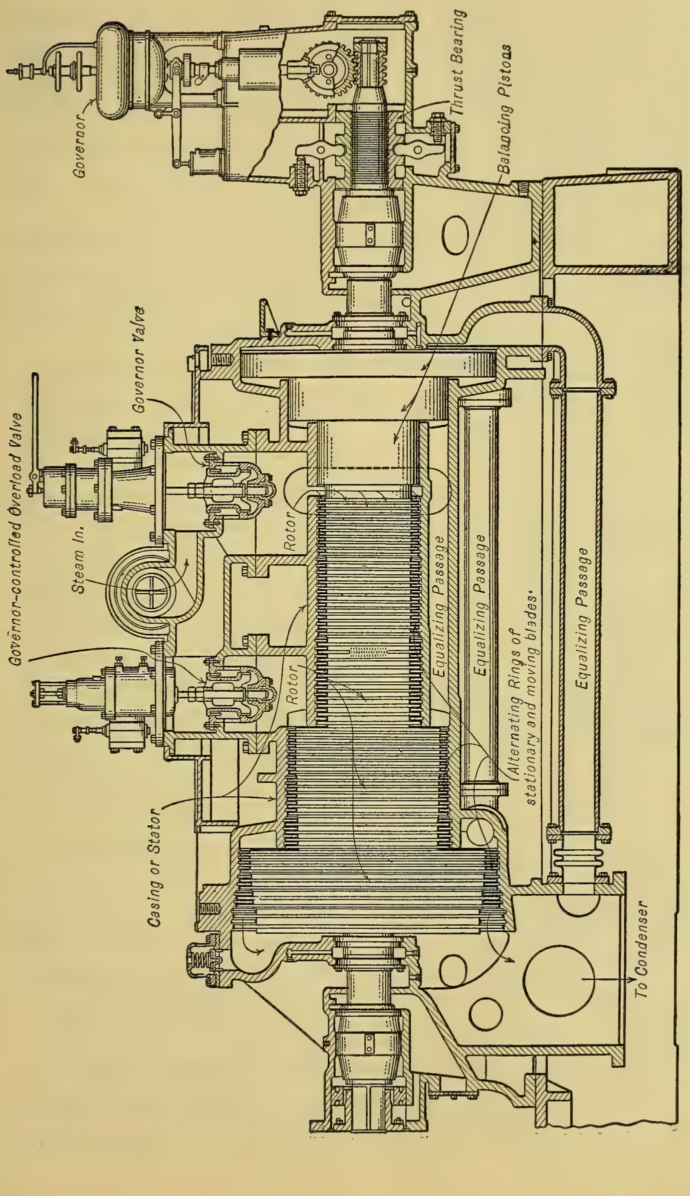


FIG. 166.—Westinghouse-Parsons Steam Turbine.

there must be a higher pressure on one side of the wheel than there is on the other. Since there must also be mechanical clearance between the blade tips and the interior of the casing, it follows that the reaction type will be handicapped by considerable leakage which does not exist in the impulse type, excepting as some of the jet may "spill" over the ends of the blades in the latter.

The difference of pressure on the two sides of the wheel also causes a tendency toward motion of the wheel along the shaft, or of the wheel and shaft, in a direction away from the higher pressure.

Many unsuccessful efforts have been made to design efficient reaction turbines, but no pure reaction type has yet been commercialized. The turbines commonly called reaction turbines are really combinations of reaction and impulse types.

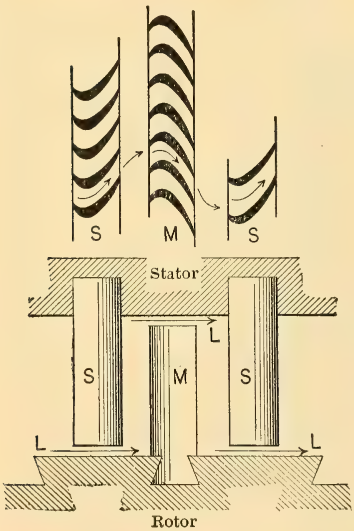


Fig. 167.

One example of what is commercially called a reaction turbine is shown in Fig. 166. Alternate rings (or rows) of stationary and movable blades guide the steam as it expands from the high pressure at one end to the low pressure at the other. The stationary blades project inward from the interior surface of the stationary casing and the movable blades project outward from the external surface of the cylindrical rotor. The rotor blades act like those of an impulse turbine in partly reversing the direction of jets of steam which reach them with comparatively high velocities, but they also act

like the movable nozzles of a reaction turbine since the steam in passing through them expands and acquires kinetic energy, giving a reaction on discharge. The stationary blades serve to redirect the steam so that it strikes the next set of moving blades at the proper angle and they also serve as nozzles in which velocity energy is acquired. This is shown diagrammatically in Fig. 167, in which S denotes stationary, and M movable blades.

The Parsons type, illustrated in Fig. 166, may be described as a *multistage type* in which impulse and reaction are utilized in conjunction.

The balance pistons shown in the figure are used to balance the end thrust caused by the difference in pressure existing on opposite sides of the wheels in the case of reaction turbines. Each piston is of such a diameter that it presents a surface equal to the blade surface acted upon by one of the unbalanced pressures, and by connecting across as shown in the figure a high degree of balance is secured.

The overload valve is used to admit high-pressure steam to the low-pressure blades for carrying excessive overloads. The larger area of the passages through these blades permits an abnormal amount of high-pressure steam to pass, thus giving a high load-carrying capacity with decreased economy.

107. Combined Types. The clearance at the ends of the stationary and moving blades in the Parsons type of turbine permits considerable steam to leak by, as previously explained. This clearance must have almost the same length (measured from blade tip to opposing metal) in all stages in order to insure freedom from rubbing, but it is more detrimental in the high-pressure stages than in the low. The high-pressure blades are much shorter than the low-pressure blades and a leakage length of a certain amount is therefore equal to a greater fraction of the total blade length. The density of the high-pressure steam is also so

much greater than that of the low-pressure steam that many more pounds can leak through an opening of a given size in a given time. In discussions of this character, it should not be forgotten, however, that leakage area is determined by the dimension already referred to multiplied into a circumference and that the circumference is much greater at the lower end.

Because of these and other reasons many manufacturers have come to the conclusion that the impulse type is best for the high-pressure end of the turbine and the reaction type for the low-pressure end. Many such combinations have been produced and they are giving very good results.

108. Steam Consumption of Steam Turbines. It is exceedingly difficult to compare the steam consumption of turbines and reciprocating engines in a general way. Roughly the steam consumption of the better varieties of the two types is of the same order for comparable conditions with the advantage probably slightly in favor of reciprocating engines in the smaller sizes and in favor of turbines in the larger sizes.

It has been shown that the steam turbine operates on the complete expansion cycle while the reciprocating engine operates on a cycle with incomplete expansion. The turbine therefore has a certain theoretical advantage because its cycle is such as to convert into work a greater amount of heat per pound of steam used between given upper and lower pressures.

This choice of different cycles rests on a sound foundation. This can be appreciated best after studying Fig. 168 which shows the volumes assumed by steam expanded adiabatically from an initial pressure of 150 pounds gauge and 100° F. superheat. It will be observed that at the lower pressures the volume increases very rapidly with a small drop of pressure. If an attempt were made to expand steam in a reciprocating engine down to exhaust pressure the cylinder would have to be increased in size by a very large amount in order to accommodate the rapidly increas-

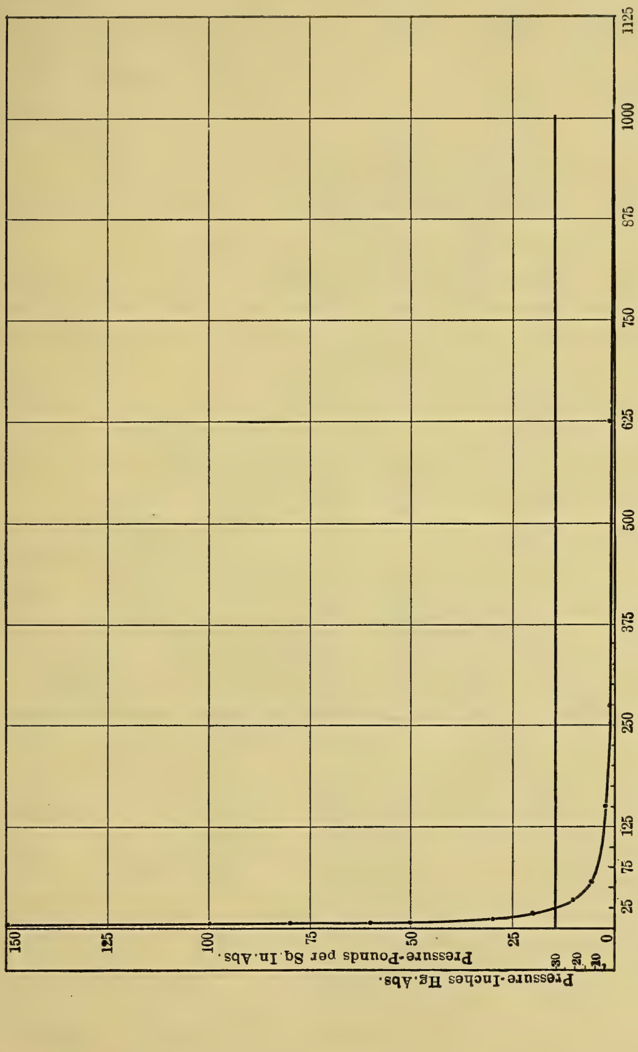


Fig. 168.—Volumes, Cu. Ft. per Pound of Steam Expanded Adiabatically from 150 Lbs. and 100° F. Superheat.

ing volume of steam. As previously stated in Section 40, the actual amount of additional energy recovered from the steam would not be worth enough to balance the increased friction loss of the larger parts and the increased investment in the larger engine. Further, some pressure difference is required to cause the steam to flow through the comparatively restricted exhaust ports and valves in the short time available and absolutely complete expansion is thus really impossible even if it were desirable.

With the steam turbine, experience has shown that it is often economical to expand the steam down to a low back pressure and as there are no restricted exhaust ports and valves no pressure differential is required. This is of great importance as a very small pressure drop at low pressures makes available a very large amount of energy in comparison with the result of a corresponding pressure drop at higher pressures.

For example, trial on a $T\phi$ or Mollier Chart will show that saturated steam in expanding with constant entropy from 200 pounds absolute to 0.5 pound absolute makes available almost as much energy in dropping from 15 pounds to 0.5 pound as it does in dropping from 200 to 15. Further, such trial will show that at the extremely low pressures a very small pressure drop liberates a relatively tremendous quantity of energy.

As a result of these characteristics of steam, the turbines which are built to give low steam consumption are designed to operate with very low back pressures, that is very "high" vacuums. In the larger sizes this introduces real difficulties in design. The entire volume of steam at the lowest pressure has to flow through the last set of blades and these must be extremely long or carried on a wheel of large diameter, or both, in order to give the necessary area for passage of the steam. Such difficulties have led to numerous "double flow" designs in which the steam is expanded from initial pressure to some lower pressure in the ordinary way

and is then introduced into a section in which it can divide and flow both ways through two opposed but similar sets of blades. One double-flow arrangement, as applied to an impulse turbine, is shown diagrammatically in Fig. 169

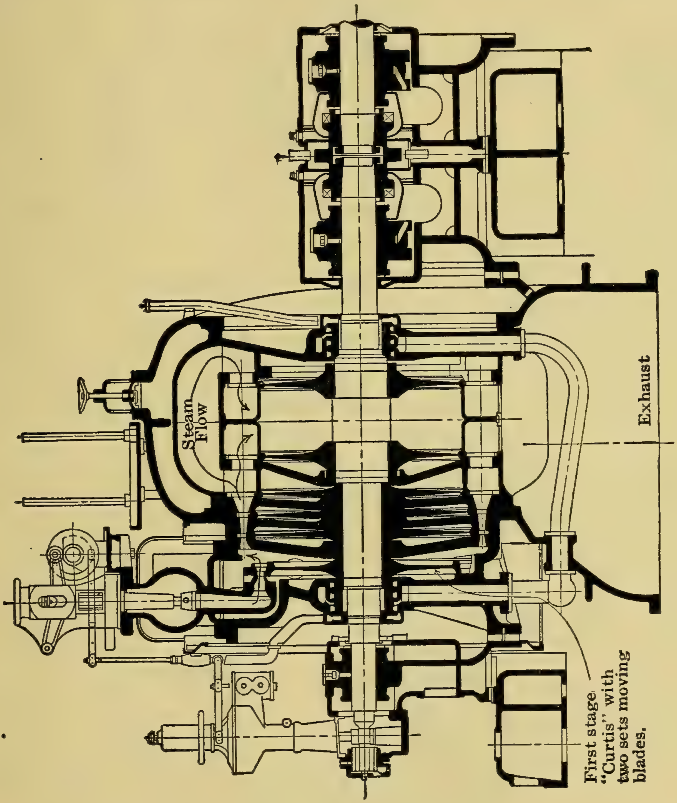


FIG. 169.—Double-flow Arrangement as Applied to an Impulse Turbine.

and another, applied to a reaction turbine, is shown in Fig. 170.

In some of the larger sizes of turbines the unit has been broken up into two or more parts. Each part is a complete turbine but is built for a smaller pressure range than that

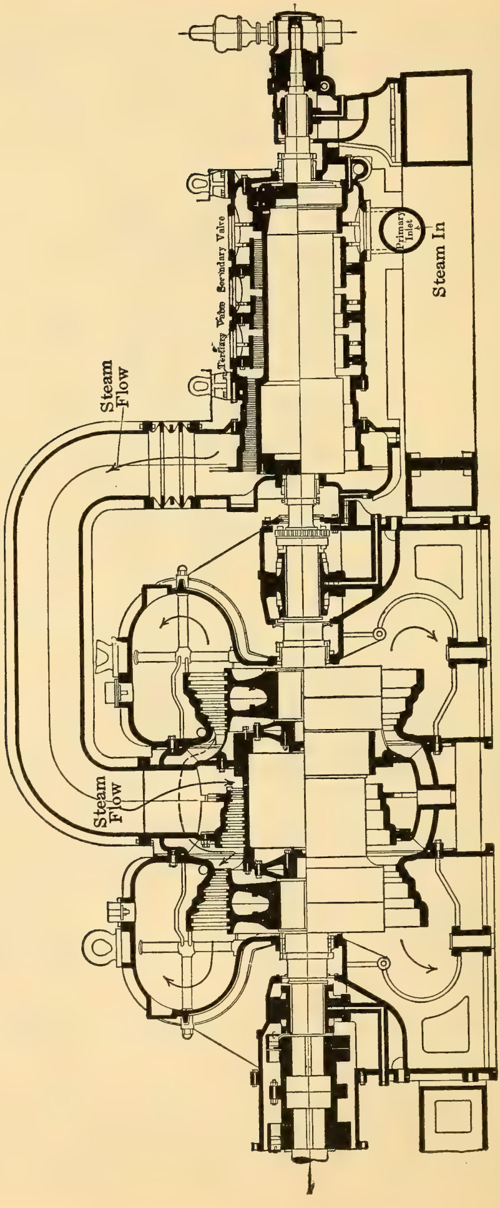


FIG. 170.—Double-flow Arrangement Applied to a Reaction Turbine.

through which the steam is expanded by the complete unit. The "high pressure unit" receives steam from the boilers and expands it down to some intermediate pressure. It exhausts into one, or two, "low pressure units" which expand the steam down to the lowest pressure and discharge it to the condenser.

This arrangement is similar to compounding as applied to reciprocating engines but it has an entirely different purpose. Compounding of turbines in this way offers greater flexibility of design. The designer can choose different rotative speeds for the high and low pressure units instead of having to use some sort of compromise as is necessary when all elements are carried on a common shaft. Compounding offers another advantage which is of importance in the larger sizes. The size and complexity of each of the parts is less than if the entire turbine were built in a single unit and it is logical to assume that this adds to the operating reliability of extremely large units, if all other features are alike.

The very real thermal advantage to be gained by using low back pressures with steam turbines is shown by results of tests which indicate that lowering the back pressure by one inch of mercury will increase the economy by from 3 to 10 per cent, depending upon the type of turbine, the back pressure under consideration and upon other factors.

Superheat is also very effective in improving the thermal efficiency of the steam turbine. In general, every ten degrees of superheat causes a saving of 1 per cent in the weight of steam required for a given output.

109. Low Pressure Turbines. Experience has shown that reciprocating engines are fully the equal of turbines in the high pressure ranges, in many cases they are even superior. The turbine, on the other hand, has the advantage at low pressures and in cases where great ratios of expansion are used. It was at one time suggested that

these characteristics should be recognized by building mixed plants, using reciprocating engines for the first part of the expansion and exhausting these into turbines at or near atmospheric pressure.

Under ordinary conditions this is not an economical solution as the investment is so high that any thermal gain which can be obtained is not sufficient to balance the increased capital charges to say nothing of increased complications. However, the scheme has been used to advantage in increasing the capacity and thermal efficiency of reciprocating plants which were already installed.

The ability of the turbine to handle low pressure steam to advantage has given rise to the use of low pressure turbines in many different ways. As examples, the exhaust of hoisting engines, steam hammers, and other apparatus commonly exhausting at atmospheric pressure is now frequently led to one or more low pressure turbines in which it is expanded with the recovery of a large amount of power from what would otherwise be waste steam.

When low pressure turbines are used in this way it frequently happens that the demand for steam on the part of the turbine and the make of steam on the part of the primary user are so different from instant to instant that some device must be used to store steam between the two. The device used for this purpose is known as a regenerator. It consists of some sort of closed vessel in which steam can be mixed with and condensed in hot water. When the make exceeds the demand of the turbine the pressure and temperature within the regenerator rise; when the demand of the turbine exceeds the make the pressure and temperature within the regenerator fall.

110. Steam Turbo-generators. The steam turbine is ideally suited to the driving of electric generators of the alternating current type as the desirable speeds for the two devices fall in the same general range of values. This fact,

coupled with the ease with which such units can be constructed in large capacities, the comparatively low cost and the high thermal efficiency attainable, has resulted in

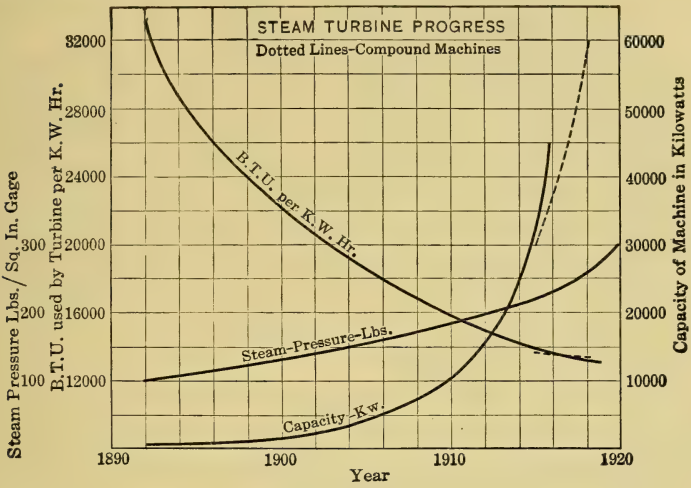


FIG. 171.—Curves Showing Progress of the Steam Turbine.

bringing the steam turbine into almost universal use for this purpose where steam is used as the means of producing power.

The history of the development of the large Turbogenerator is shown in Fig. 171.

PROBLEMS

1. A steam turbine produces one horse-power hour at its shaft for every 30 lbs. of steam supplied. The initial pressure is 200 lbs. absolute and the steam is superheated 200° F. The turbine exhausts against a back pressure of 14 lbs. absolute.

Find the thermal efficiency on the assumption that heat of liquid at exhaust temperature is not chargeable to the turbine.

2. Develop a complete expansion cycle for one pound of material used under the conditions of Prob. 1 and find the energy

made available per cycle. From this value determine the number of pounds of material theoretically required per horse-power-hour and compare with the value given in Prob. 1.

3. Find the additional quantity of energy which would theoretically be made available per pound of steam in above problems if the back pressure could be lowered to $\frac{1}{2}$ lb. absolute.

4. Develop a complete expansion cycle from an initial pressure of 225 lbs. absolute with a superheat of 200° F. to a back pressure of $\frac{1}{2}$ lb. absolute. Assume that this is to be divided up into six parts, each making available the same quantity of energy. Find the pressure drop for each part. Note that this is most easily done with the help of the Mollier chart.

5. A steam turbine receives steam at a pressure of 225 lbs. per square inch absolute and with a superheat of 190° F. and exhausts into a condenser in which a pressure of $\frac{3}{4}$ lb. per square inch absolute is maintained. The turbine is direct connected to an electric generator and produces a K.W.-hour on 12 lbs. of steam. If a K.W.-hour is equivalent to 3411 B.t.u., what is the thermal efficiency of the combination?

6. Develop a complete expansion cycle for the conditions of Prob. 5 and determine the pounds of steam which would be required theoretically to develop energy equivalent to 1 K.W.-hour. Compare with the value given in Prob. 5.

7. Determine the velocity theoretically attainable by expanding steam in one step from the initial to the final conditions of Prob. 5 above. What would be the value of the kinetic energy of such a jet per pound of steam flowing?

8. Determine the shape of a nozzle required to discharge 1000 lbs. of steam per hour, initial conditions being 100 lbs. per square inch absolute, and dry saturated steam; final pressure being 2 lbs. absolute.

9. Determine velocity and kinetic energy of jet in Prob. 8.

CHAPTER XIV

CONDENSERS AND RELATED APPARATUS

111. The Advantage of Condensing. The lowest pressure to which an engine or turbine can expand steam, that is the exhaust pressure, is determined by the pressure prevailing in the space into which the steam is exhausted. With a given initial pressure the amount of work which can be obtained theoretically from a given weight of steam increases as the exhaust or back pressure decreases, as shown by the areas of the two diagrams in Fig. 172, and experience has shown that at least a part of this theoretical increase can be obtained in real engines. It is therefore desirable to exhaust into a space in which the lowest possible pressure exists when the work obtained per pound of steam is the only consideration.

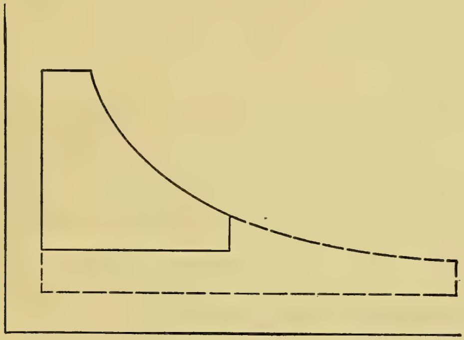


FIG. 172.

The most available space into which an engine can

exhaust is that surrounding the earth and already occupied by the earth's atmosphere. The pressure in this space is approximately equal to 14.7 lbs. per square inch at sea level and is due to the weight of the atmosphere. Since the superincumbent column of atmosphere decreases in depth as one moves upward, its weight also decreases and atmospheric pressure therefore averages less than 14.7 lbs. per square inch at high altitudes and has a greater average value at points below sea level.

If it is desired to exhaust into a pressure lower than atmospheric a means of maintaining such an abnormal pressure within some sort of vessel must be devised. It is the purpose of a condenser and its associated apparatus to make available a space in which such a low pressure can be maintained. Its method of operation will be considered in later sections.

There is also another advantage which may be obtained by the use of a condenser. It often happens that the water available is not well adapted to use in boilers. It may be salt water as in marine practice, or it may contain a number of undesirable gases and solids in solution as often occurs in stationary practice. Some types of condensing apparatus are so arranged that the steam exhausted from the engine is converted into liquid water without admixture and can therefore be returned to the boiler as practically pure water, thus largely eliminating the troubles that would ensue from the use of poor feed water.

112. Measurement of Vacuum. Assume that some non-volatile liquid, that is, a liquid that did not vaporize, could be found and also that it contained no gases in solution. If a long tube were inserted in a vessel filled with such a liquid and had its upper end connected with some form of vacuum pump which could remove air from its interior, as shown in Fig. 173, liquid would rise in the tube as the air was removed. Removal of air would result in lowering the pressure within the tube, but the constant

atmospheric pressure on the liquid surface outside the tube would then force liquid up the tube to such a height that the pressure p_a of air still in the tube plus the pressure due to the column of liquid of height h within the tube just equaled the pressure due to the atmosphere on the surface of the liquid in the vessel. If the pump could remove all of the air from the tube, liquid would rise to such a height that the pressure exerted by it on a plane passing through the lower surface just equaled that of the external atmosphere.

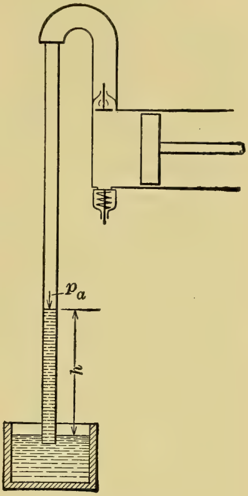


FIG. 173.

The same result could be attained by using a tube closed at one end, filling it with the liquid, and then inverting so that the end rested in the liquid as shown in Fig. 174. If the tube were long enough, the liquid would drop to some such point as shown, under which conditions the height of liquid would just balance atmospheric pressure. This would only be true if the liquid did not volatilize and did not contain gases in solution; with these assumptions the space above the liquid in the tube would contain absolutely nothing. This space would be said to be perfectly vacuous, or a perfect *vacuum* would be said to exist in that part of the tube.

FIG. 174.

A device of this character is used to measure the pressure of the atmosphere and is known as a **barometer.** Mercury is used as the liquid because its high density makes it possible to use a short tube and because it may be considered

as non-volatile at ordinary temperatures. The average atmospheric pressure at sea level, equal to 14.7 lbs. per square inch approximately, can support about 30 ins. of mercury, so that this figure is generally taken as the standard sea level barometer reading. An atmospheric pressure of less than 14.7 lbs. would be shown by a barometer reading of less than 30 ins.; a greater atmospheric pressure by more than 30 ins. Corresponding values of atmospheric pressure and barometer reading are given in Table VII. To this have also been added the altitudes to which the different values would correspond if a pressure of 14.7 lbs. existed at sea level and there were no variations of atmospheric pressure excepting those due to change of elevation. Values of this type can only be roughly approximate, because local barometric variations are constantly occurring and the sea-level atmospheric pressure varies both sides of 14.7 lbs.

TABLE VII

Atmospheric Pressure, Barometer Reading and Altitude

(Negative signs mean distance below sea level.)

Barometer, Inches of Mercury.	Atmospheric Pressure, Pounds per Square Inch.	Altitude, Feet (Approximate).
25.00	12.27	4750
26.00	12.76	
26.50	13.01	3250
27.00	13.25	
27.50	13.49	2250
28.00	13.74	
28.50	13.98	1300
29.00	14.23	
29.25	14.35	
29.50	14.47	450
29.75	14.60	
30.00	14.72	Sea level
30.25	14.84	
30.50	14.96	−450
30.75	15.09	
31.00	15.21	−900

The exact value of standard atmospheric pressure at sea level is taken at 29.921 ins. of mercury, which is equal to 14.696 lbs. per square inch and corresponds to the 760 mm. of mercury, used by scientists as standard.

A tube with both ends open and arranged as shown in Fig. 173 can be used to measure the degree of vacuum existing in the space to which its upper end is connected, and many vacuum gauges are constructed on this principle, using mercury as the liquid. The extent to which the pressure is lowered in the top of the tube is indicated by the height to which the mercury column rises and this height in inches is used as a measure of the vacuum. Thus if a perfect vacuum were created and if the atmospheric pressure were equal to 14.7 lbs. the gauge would show about 30 ins. of mercury above the level in the reservoir. If the vacuum were less perfect the gauge would show a shorter column.

It should be noted that the reading of the vacuum gauge does not give the pressure existing in the vacuous space, but gives the amount by which the pressure has been reduced below that of the atmosphere, the difference being expressed in inches of mercury. By subtracting this reading from the existing atmospheric pressure expressed in the same units, the *absolute pressure* in the partially vacuous space (expressed in inches of mercury) is obtained.

It is obvious, therefore, that a vacuum-gauge reading of say 28 ins. of mercury does not always mean the same absolute pressure. With a barometer reading of 28 ins. it would represent a perfect vacuum; with a barometer reading of 30 ins. it would represent a partial vacuum, the absolute pressure in the partially vacuous space being equal to 2 ins. of mercury.

113. Conversion of Readings from Inches of Mercury to Pounds per Square Inch. It is often necessary to convert readings of pressure in inches of mercury into pounds per square inch. This can be done with sufficient accuracy

under ordinary circumstances by multiplying the inches of mercury by the constant 0.4908. Thus,

$$\text{Barometer in inches} \times 0.4908 = \text{atmospheric pressure in pounds per square inch} \quad . \quad . \quad . \quad (74)$$

and

$$(\text{Barometer in inches} - \text{vacuum gauge in inches}) \times 0.4908 = \text{absolute pressure in partially vacuous space in pounds per square inch.} \quad . \quad (75)$$

ILLUSTRATIVE PROBLEM

A vacuum gauge constructed like that shown in Fig. 167 reads 27 ins. when the barometer reads 29.5 ins. What is the absolute pressure in the partial vacuum above the mercury?

The absolute pressure is equal to $29.5 - 27 = 2.5$ ins. of mercury, which is equal to

$$2.5 \times 0.4908 = 1.227 \text{ lbs. per square inch.}$$

114. Principle of the Condenser. A perfect vacuum could be created in any closed vessel with impenetrable walls if a pump could be devised which could remove all material contained within that vessel. Or, any degree of vacuum can be maintained in any partially closed vessel by fitting to it a pump which can remove all material flowing into the vessel as fast or faster than it enters, raise the pressure of this material to atmospheric or higher and discharge it.

The latter principle is made use of in real condensers, a pump of some form, or an equivalent, removing from the condenser the material exhausted by the engine and inleakage from the atmosphere, and discharging it at atmospheric pressure at a sufficiently rapid rate to maintain the desired vacuum. If the condenser and connections could be made leak proof, the pump or equivalent would have to handle only the material exhausted from the engine.

A steam engine exhausts a mixture of steam, water and gases, the gases being a mixture of those originally

dissolved in the boiler-feed water and air which leaks into those parts of the system in which a partial vacuum is maintained. If the pump had to handle the same volume of material as is exhausted by the engine, no gain of work would result from condensing, because the pump would have to do at least as much work in raising the pressure of this material to atmospheric and discharging it as could be obtained by allowing it to expand in the engine.

Steam, however, occupies a much larger volume than water at the same temperature and pressure. Thus steam at 212° F. occupies a volume of about 26.79 cu.ft. per pound, but water at the same temperature and pressure occupies a volume of only about 0.0167 cu.ft. per pound; at a temperature of 120° F. which is often used in condensers, the specific volume of steam is about 203 and that of water only 0.0162. Therefore, if the steam is condensed after exhaust from the engine and before entering the pump to be discharged to atmosphere, the pump work is greatly reduced. The volume of the condensate is almost negligible in comparison with the volume of steam exhausted, and the work of pumping it is almost negligible in comparison with the work it made available in the engine.

Gases contained in the exhaust steam cannot be liquefied and must be pumped as gases. The work required to pump them can, however, be reduced by lowering their temperature as far as possible.

The condenser equipment may be regarded as consisting of a combination of a partially closed vessel and some form of pump. The vessel is so constructed that a low temperature can be maintained within it and that large quantities of heat can be removed from it for the purpose of condensing the exhaust steam and of cooling the contained gases. This is generally done by using large quantities of cool water.

The absolute pressure within the condenser is made up of two parts. The two parts are, (*a*) that due to the

water vapor, since the space over the condensed water will always be filled with saturated steam at the same temperature (approximately) as that of the water, and (*b*) that due to any gases present.

The pressure of the saturated steam (water vapor) can be found from the steam tables opposite the temperature existing in the condenser and it is the pressure which would exist in the condenser of an ideal system in which no gases were mixed with the working substance. The pressure of the gases can be found by subtracting from the total measured pressure in the condenser the pressure exerted by the water vapor as shown in the steam tables. The pressures exerted by the water vapor and gases are spoken of as partial pressures, since their sum makes up the total pressure within the condenser.

The presence of gases causes a two-fold loss. First, it increases the pressure against which the engine has to exhaust, thus raising the back-pressure line on the diagram and decreasing the work area. Second, it increases the work which must be done by the pump which otherwise would only pump the condensate and such saturated water vapor as accompanied it.

115. Types of Condensers. The condensers actually used in steam plants can be roughly divided into two types, as

(*a*) *Contact condensers* and

(*b*) *Non-contact condensers.*

In the first type the water which is used for condensing and cooling is intimately mixed with the exhaust from the engine within the condensing vessel, and the resultant mixture of condensing water, condensate and gases is drawn out of this vessel and discharged to atmosphere by the pump.

In the second type condensing water flows on one side of metal surfaces of some sort and the exhaust is led over the other side, the heat being transmitted through the

metal. In condensers of this type the condensate and gases are not mixed with the condensing water and the condensate can therefore be returned to the boiler as feed water with the advantages already mentioned.

116. The Jet Condenser. One of the earliest forms of contact condensers which is still very widely used for moderate vacuums is commonly known as the *jet condenser*. The principle of operation of the jet condenser is shown in Fig. 175. Water, under pressure, entering as indicated, is broken up into fine streams or jets and sprayed into the exhaust coming from the engine. The resultant mixture flows downward into the neck of the condensing vessel or "condenser head" and is removed by some form of pump. This pump handles gases, vapors and water and is known as **a vacuum pump**, *a wet-vacuum pump*, or *a wet-air pump*, the term wet signifying that it handles the water as well as the gases.

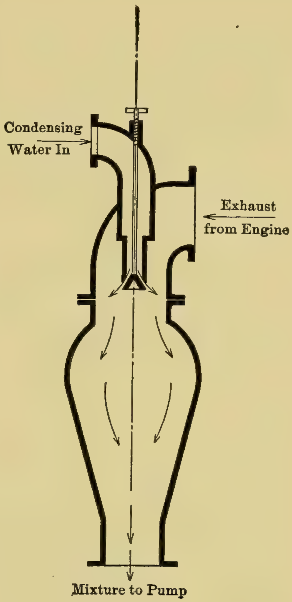


FIG. 175.—Jet Condenser.

The pressure within such a condenser head would be theoretically equal to that corresponding to the temperature of the resultant mixture if no gases were present. In practice the pressure of the water vapor would roughly correspond to the average temperature near the top of the vessel and there would be a partial pressure due to gas as well. This gas would consist of that brought over by the engine exhaust plus that released from the cooling or "circulating" water under the low pressure within the condenser.

Details of a complete jet condenser and of the method of connecting it to an engine are given in Fig. 176. The atmospheric relief valve is installed in all condensing systems and is arranged to open automatically and thus discharge exhaust steam to atmosphere if the pressure within the system rises to atmospheric, that is, if the "vacuum is lost."

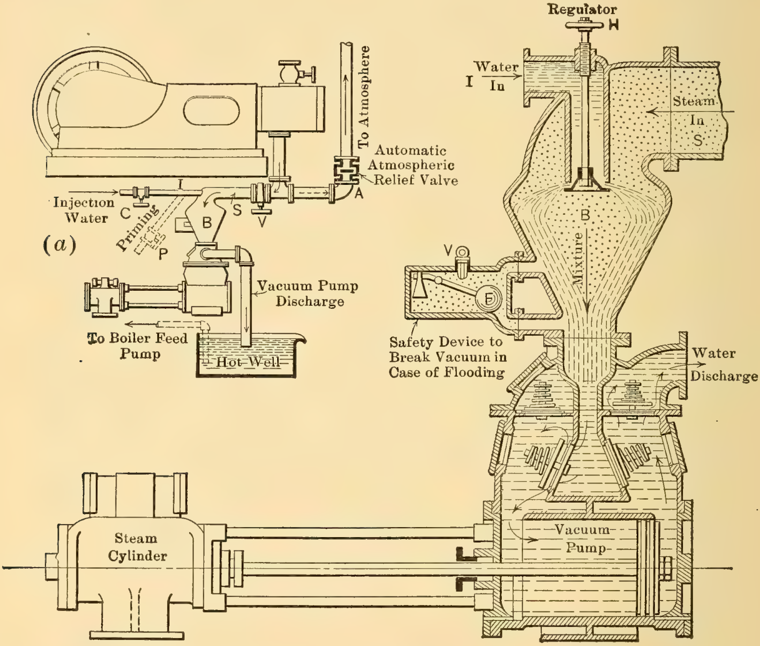


Fig. 176.—Jet Condenser and Method of Connecting to Engine.

With the jet condenser the pressure might start to rise because of slow action or even stoppage of the pump. As the condenser head filled up the rising water would ultimately entirely cover the jet and condensation would then practically cease. In the arrangement shown in Fig. 176 there is an additional safety device which breaks the vacuum in the exhaust system if the water in the head

rises above a certain height, thus preventing the external atmospheric pressure from forcing this water back along the exhaust pipe and into the cylinder, an event which would probably result in a wrecked engine.

The jet condenser here described is known as a **parallel-flow type,** because everything within the condensing vessel flows in the same direction. The gases and vapors handled by the pump theoretically have the same temperature as that of the mixture with which they flow out at the bottom of the condenser head. The temperature of this mixture therefore determines the temperature of the gases and vapors pumped.

There are numerous forms of contact condensers which more or less closely resemble the types of jet condenser just described. They are properly all classed as jet condensers, but more often are given distinguishing names.

One very common form of contact condenser is generally known as a **barometric condenser.** It consists essentially of a condenser head, similar to that used with the jet condenser already described, and a tail pipe or barometric pipe which partly or wholly takes the place of the wet-vacuum pump by removing part or all of the mixture formed within the condenser. One model of such a condenser is shown in Figs. 177 and 178.

The exhaust from the engine enters the head through the large pipe shown and divides into two parts, one part passing down through the center of the head and the remainder flowing downward in the annular space *A*. The circulating or injection water enters as shown, is divided by the spraying cone and injected into that part of the exhaust, which enters the central tube of the condenser. The mixture thus formed flows downward and finally meets the discharge from the lower end of the annular space *A*, which is then condensed. The mixture of injection and condensing water together with such gases as have been entrapped, then flows downward into the tail pipe,

which is over 34 ft. in length and which dips into the "hot well" at its lower end. As atmospheric pressure can only

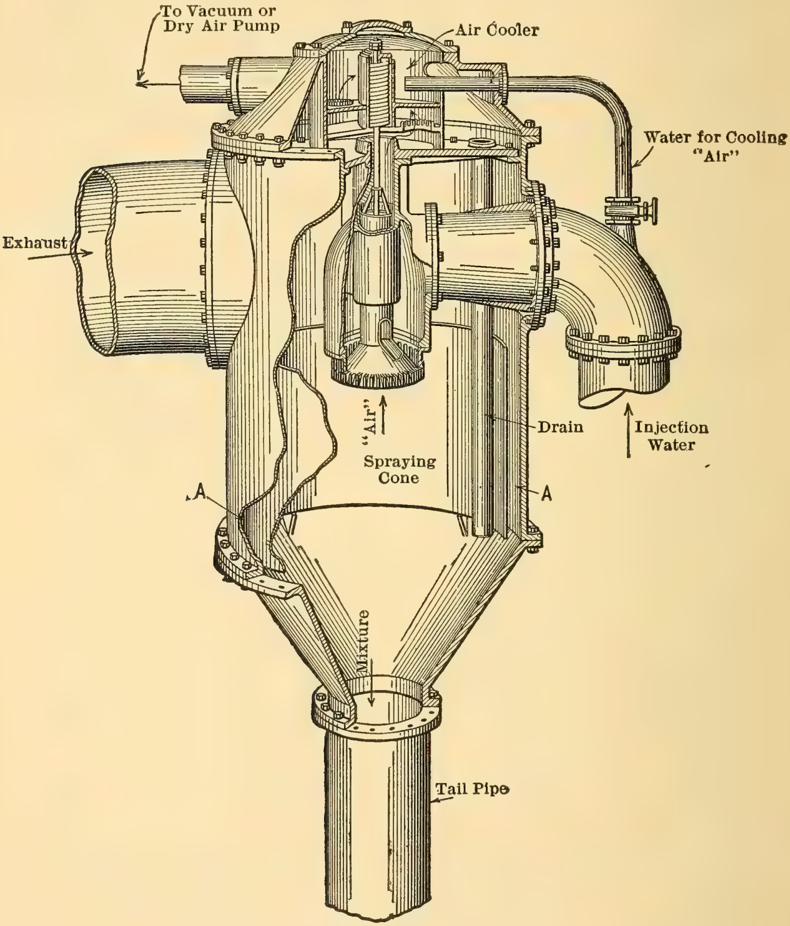


FIG. 177.—Barometric Condenser.

support a column of water about 34 ft. high, the tail pipe forms an automatic wet-vacuum pump, water flowing from it as rapidly as it accumulates within it.

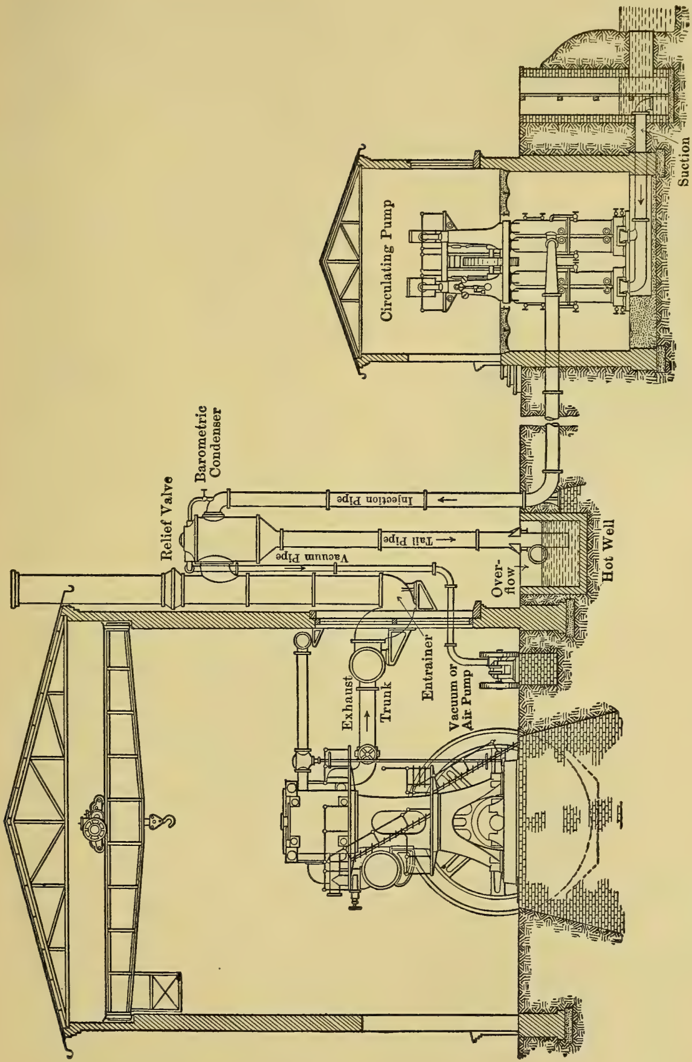


FIG. 178.—Method of Connecting Barometric Condenser to Steam Engine.

Experience has shown that the maintenance of a high vacuum with this type of condenser depends upon the extent to which gases are removed from the condenser head. These gases are generally called air, as the greater part of them is air. In the type illustrated such "air" as is not trapped by the descending mixture rises through the hollow spraying cone, then through the air cooler and flows out through the pipe indicated to the vacuum or dry-air pump. The air in rising through the center of the spraying cone is cooled by the water flowing around it, and it is further cooled by coming into contact with water as it works its way through the air cooler. This results not only in lowering its temperature, but also in causing the condensation of a great deal of the water vapor accompanying it. This condensed vapor collects in the space surrounding the air cooler and flows down into the head through the drain shown. The vacuum pump, therefore, handles cool gases containing little water vapor and practically no liquid water. It is sometimes called a **dry-air pump** or **dry-vacuum pump** for this reason.

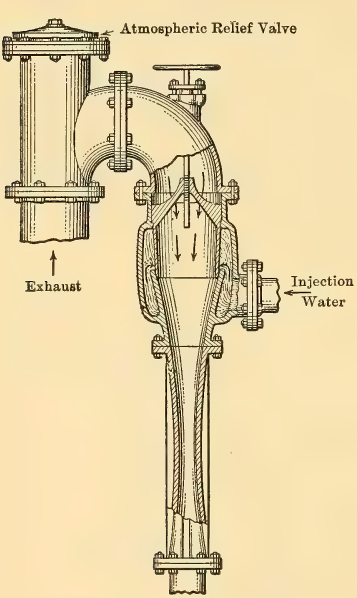


Fig. 179.—Baragwanath Barometric Condenser.

The entrainer shown in the exhaust system in Fig. 178 is so shaped that water collecting in the exhaust piping and flowing into the entrainer is picked up by the exhaust steam and carried into the condenser.

The flow of steam and injection water in this condenser is parallel, but the material on its way to the dry-vacuum pump flows upward and the cooling water flows downward so that *counter-current flow* is used in this part of the apparatus.

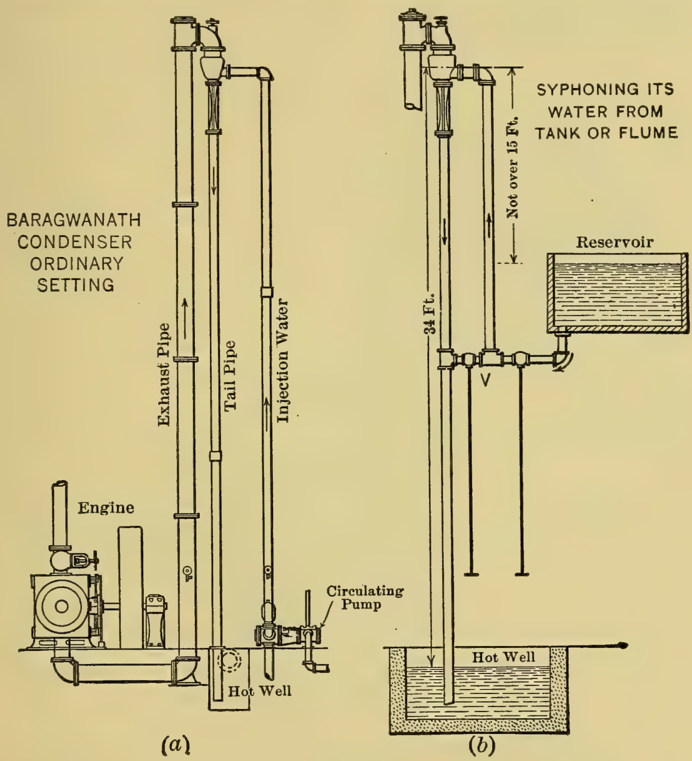


FIG. 180.

ratus. This has the advantage of bringing the *air* leaving the condenser into contact with the cooling water just as it enters and therefore when it has its lowest temperature.

A somewhat similar condenser, arranged so that it requires no pump, is shown in Figs. 173 and 174 (*a*) and (*b*).

Exhaust and injection water mix as shown, the quantity of injection water being regulated by the hand wheel on top of the condenser. The mixture flows downward through the narrow neck and the velocity attained in this part of the tail pipe is so high that all air and similar gases are swept along with the current.

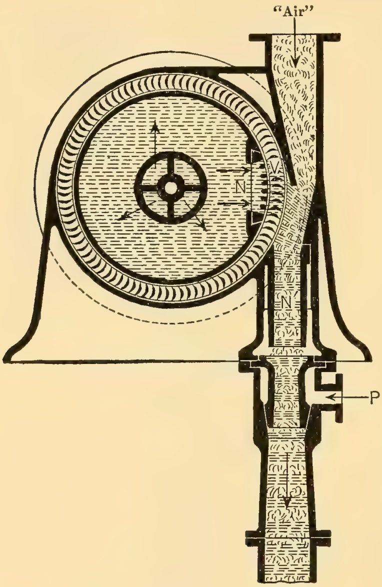


Fig. 181.—Westinghouse-Leblanc Air Pump.

Fig. 180 (*b*) shows an arrangement not requiring a circulating pump. For starting, the valve *V* is opened, allowing water to flow into the lower part of the tail pipe. This creats a partial vacuum, and atmospheric pressure then forces water up the injection pipe and into the condenser head. The valve *V* is then closed and the condenser continues to siphon its own water. Because of this action this type is often called a **siphon condenser.** By supplying a circulating pump as indicated in Fig. 180 (*a*) it can be converted into a barometric condenser similar to the type already discussed except for the fact that it requires no air pump.

The barometric or tail pipe of any barometric condenser can be replaced by any kind of a pump, and centrifugal pumps are often used for this purpose. Centrifugal air pumps have also been devised and are in use.

A "low level jet condenser" in which the barometric tube is replaced by a centrifugal pump and in which a separate air pump of a rotary type is used is illustrated in Figs. 181, 182 and 183. It consists essentially of the condensing head and well, combined with a centrifugal tail pump and a rotary air or vacuum pump as indicated in Fig. 183.

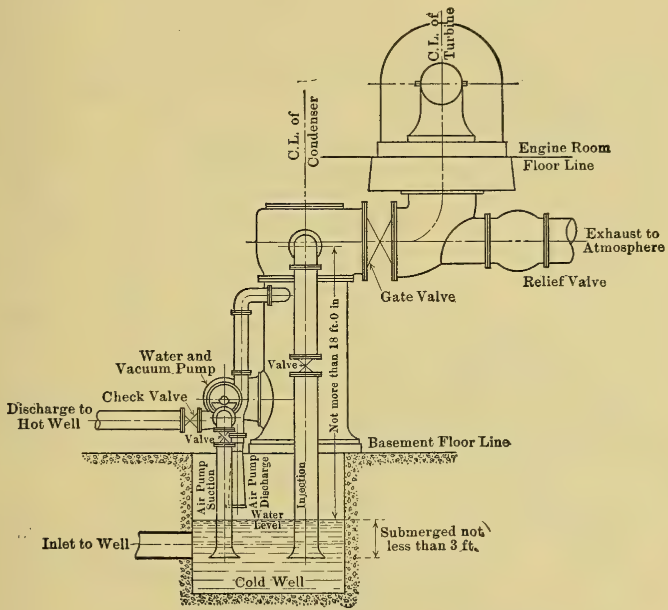


FIG. 182.—Westinghouse-Leblanc Condenser.

Injection water entering through nozzles in the head meets the exhaust, and the resultant mixture flows down into the well through the large nozzle shown. The liquid is continuously removed from the bottom of this well by the centrifugal tail pump and discharged to the hot well. The air and associated vapors are drawn down the air pipe and

discharged by means of the device shown in Fig. 181. Water enters the central part of this pump as indicated in Fig. 182 and is discharged through the stationary nozzles and the moving vanes V shown in Fig. 181. The water is thus caused to form a series of "pistons" which move rapidly downward in the discharge nozzle N' and which trap small plugs or lamina of "air" between them and thus discharge the "air" to the atmosphere. The connection marked P is used for priming at starting when necessary.

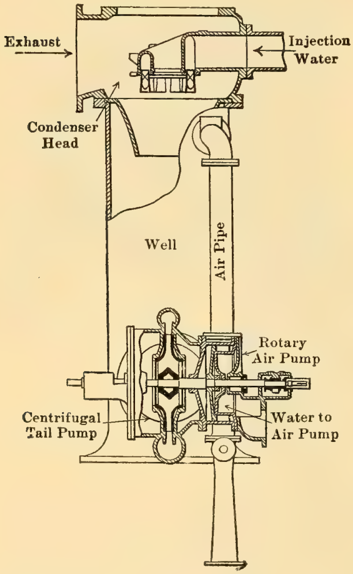


FIG. 183.—Westinghouse-Leblanc Condenser.

In small units the centrifugal tail pump may be omitted and the design so remodeled that all the injection water passes through the rotary air pump which discharges the entire mixture from the condenser just as it discharges the air and associated vapors in the larger sizes.

117. Non-contact Condensers. The type called the **surface condenser** is the best-known example of *non-contact condenser*. It consists essentially of a long vessel of cylindrical, rectangular or other section into which the exhaust is discharged and through which pass numerous brass or alloy tubes which carry the circulating water, and the surface of which acts as the condensing and cooling surface.

One form of surface condenser mounted above the pumps which serve it is shown in Fig. 184. The exhaust enters at the top of the rectangular shell and works its

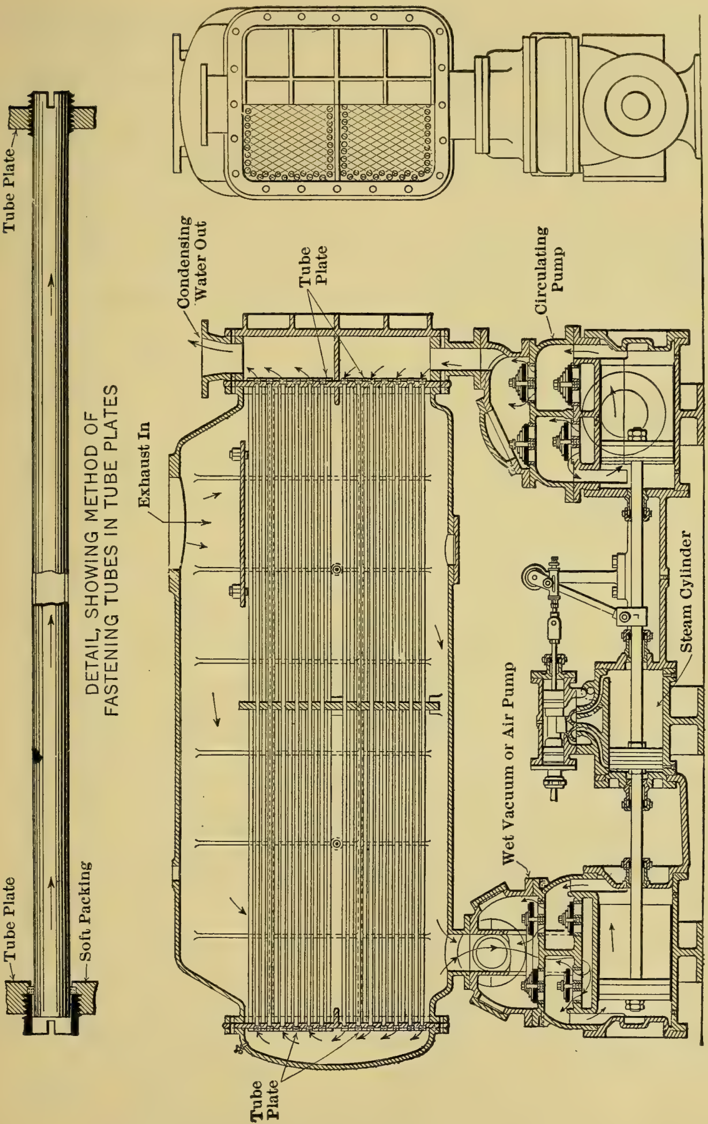


Fig. 184.—Wheeler Admiralty Type Surface Condenser.

way down over the water-cooled tubes. The condensate, mixed with gases and vapors, is drawn from the bottom of the shell by the wet-vacuum pump and discharged to the hot well.

The condensing water is forced through the tubes of the condenser by means of the reciprocating circulating pump, entering the lower tubes at the right-hand end in the figure, making two " passes " through the condenser and leaving at the top. Because of the path of the water a condenser of this type is sometimes called a **two-pass** or **double-flow condenser.**

With the arrangement illustrated, the steam which condenses upon the upper tubes falls as a rain from tube to tube until it finally settles at the bottom and is drawn off. The outer surfaces of the lower tubes are therefore practically covered with water and this has two disadvantages. First, these tubes carry the coolest circulating water and they therefore cool the condensate coming in contact with them while the water flowing through them is unnecessarily heated. Cooling of the condensate means a lower hot-well temperature than would otherwise be obtained, but if the condensate is to be used for boiler feed, the temperature of water in the hot well should be maintained as high as possible, since this water will eventually have to be heated to boiler temperature with a corresponding expenditure of heat. Second, tubes which are being used to cool water covering them are of little use as condensing surface, and hence such surface in a condenser is comparatively inactive.

The ideal arrangement would carry away the liquid condensate as fast as formed, leaving the tubes first entered by the condensing water to act as the final condensing and cooling surfaces, thus bringing gases and non-condensible vapors into contact with the coolest surfaces just before entering the vacuum pump. Numerous designs which approximate this ideal have been developed recently and

they give better results than do the earlier and simpler forms. The improvement is shown by the values of condensing surface per developed horse-power of engine. In early designs it was customary to supply $2\frac{1}{2}$ sq. ft. of tube surface or more per horse-power. Some of the most recent installations are giving better vacuums with only 1 sq. ft. per horse-power.

One of these condensers passes the condensate through a set of tubes so located that the engine exhaust strikes them before impinging on any tubes carrying condensing water. This results in a partial condensation of the exhaust and raises the temperature of the condensate within the tubes to very near that of the exhaust, thus heating the boiler feed to a temperature practically corresponding to the exhaust temperature of the engine.

Surface condensers are commonly operated with a vacuum of from 24 to 26 ins. of mercury when used with reciprocating engines and with a vacuum of 28 to 29 ins. when receiving the exhaust of steam turbines. When operated at the lower vacuums wet-vacuum pumps are generally used, but the best types of dry-air pumps must be installed in combination with well-designed condensers when the higher vacuums are sought. An installation of a surface condenser and necessary auxiliaries as applied to a steam turbine of moderate size is shown in Fig. 185. The steam exhausted by the turbine enters the top of the condenser shell and spreads out over the tubes. As it is condensed it gravitates to the lower part of the shell and finally flows into the " hot well " attached to the lower part of that shell. It is removed from the hot well by the " hot well pump " which discharges it to storage tanks or heaters, depending on the layout of the plant.

The cooling water is supplied by the " circulating pump " shown. After making two " passes " through the tubes, it flows away at the " circulating water overflow."

The noncondensible gases are drawn from an " air box "

near the bottom of the condenser by means of an hydraulic air pump which will be described later. They flow through the pipe indicated as "dry air suction," enter the "hydraulic vacuum pump," and are discharged (in intimate mixture with water) into the "sealing tank." Here the noncondensible gases separate and escape through the "sealing tank vent." The water required by the hydraulic vacuum pump is circulated by the "hydraulic supply pump" and

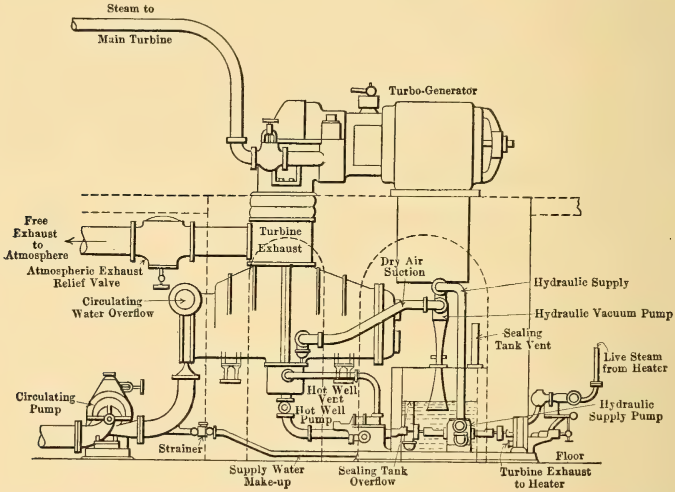


FIG. 185.—Surface Condenser and Necessary Auxiliaries as Applied to Steam Turbine of Moderate Size.

any loss is made good through the line marked "supply water make up."

In the case shown, the hot well pump and the hydraulic supply pump are driven by a single steam turbine, the exhaust of which is carried to the feed water heater and heats the condensate on its way to the boilers.

118. Vacuum Pumps or Air Pumps. In an earlier section, attention was called to the fact that some means must be provided for removing noncondensible gases, or "air,"

from condensers. This may be done by a pump which handles both the condensate and the noncondensible material. Such a pump is known as a " wet vacuum pump." A section of a pump of this kind is shown in Fig. 176 and another in Fig. 184.

For larger installations, and particularly those in which a very high vacuum is desired, it is customary to use one pump for handling the condensate and another pump for the noncondensible gases. The latter is known as a " dry vacuum pump " or more commonly, an " air pump."

The earliest dry vacuum pumps were merely reciprocating air compressors which worked between a high vacuum and atmospheric pressure. That is, they received " air " at the pressure existing in the condenser and compressed it to such a pressure that they could discharge it to atmosphere. Such pumps are still in use. They are commonly arranged with crank and fly wheel and are known as " Rotative Dry Vacuum Pumps " or " R. D. V. Pumps."

The air cylinder of such a pump is shown in Fig. 186. Air enters at the flange indicated and flows into the right-hand end of the cylinder through the upper set of valves as the piston moves to the left. On the return stroke the air is discharged without compression through the lower set of valves, and travels through a passage outside the cylinder to the space being opened up on the other side of the piston, entering through the valves shown at the left and above the bottom of the cylinder. When the piston makes its next stroke to the left, the air entrapped in the left-hand end of the cylinder is compressed and discharged through the valves at the bottom. Such a pump is described as " two stages in one cylinder." The right-hand end merely serves as a " loader " for the left-hand end and all compression occurs in the latter. This " loading " or " two stage " feature is introduced for the purpose of increasing the volumetric efficiency, that is, the amount of air handled per revolution.

With the arrangement shown, the right-hand end of the cylinder and its clearance are completely filled with air at suction pressure at each stroke. That part filling the piston displacement is air drawn from the condenser and that part filling the clearance is air left over from the previous return stroke. All of the air filling the piston displacement is

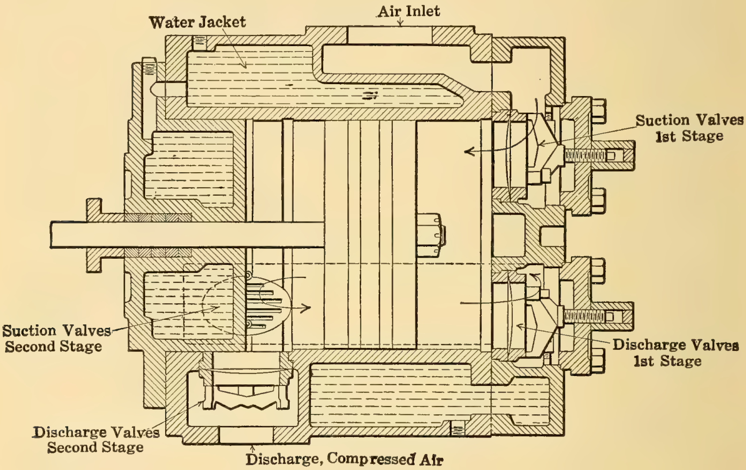


FIG. 186.—Two-stage Single Air Cylinder.

transferred to the other end for compression and discharge during the return stroke of the piston.

If the pump were designed to compress to atmospheric pressure at the right-hand end and to discharge the air to the atmosphere, its clearance would be filled with air at atmospheric pressure at the end of the discharge. When the piston moved to the left, this clearance air would have to expand to condenser pressure before any air could enter through the suction valves from the condenser. Experience shows that a relatively large part of the total piston displacement is uselessly wasted in such re-expansion of air caught in the clearance. For this reason, practically all

the better rotative dry vacuum pumps embody in their design some feature intended to minimize this effect.

After reciprocating air pumps had been developed to a point where little further improvement seemed possible,

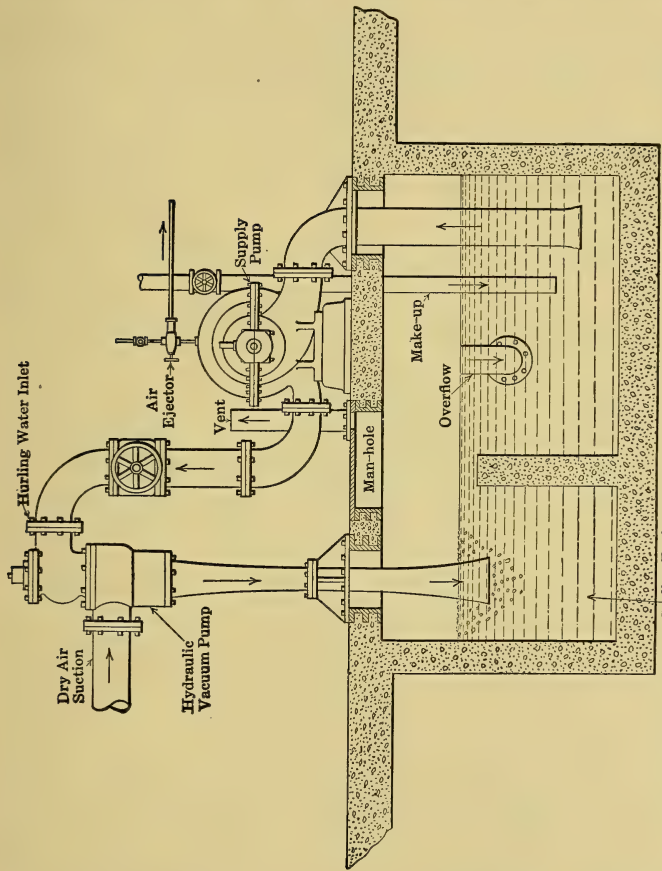


Fig. 187*a*.—Hydraulic Vacuum Pump with Supply Pump and Sealing Tank.

two radically different types of air pumps were introduced. One kind may be called the "hydraulic type" and the other the "steam jet type."

One example of hydraulic air pump is shown in Figs.

181, 182 and 183. Another, as built by the Worthington Pump & Machinery Corporation, is illustrated in Fig. 185 and is shown in detail in Fig. 187.

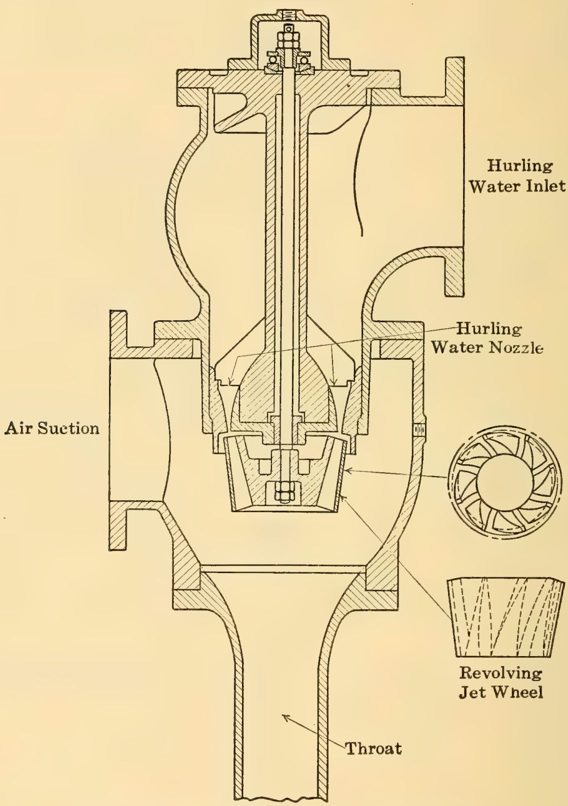


FIG. 187*b*.—Hydraulic Vacuum Pump.

In the type shown in Fig. 187, water supplied by a centrifugal pump enters the air pump at the hurling water inlet. It flows out of the head of the air pump through the annular nozzle indicated and enters the revolving jet wheel with a high velocity. The water leaves the wheel

in the form of a number of jets of approximately rectangular section, each jet traveling a helical path leading it into the throat of the discharge tube.

In traveling across the space between the revolving wheel and the throat, the jets entrap air between them and carry this air into the discharge tube. In passing through the lower part of the discharge tube, the velocity of the mixture decreases and the pressure increases so that the mixture can be discharged into the sealing tank against a small head of water plus atmospheric pressure on the surface of the water.

In the tank the air separates and escapes through the vent. The water is used over and over, any loss by evaporation or spillage being made up as convenient.

In the steam jet type, one or more steam jets entangle or entrain the condensible gases while moving at a high velocity and the mixture then passes through an expanding tube or nozzle in which its velocity is reduced with a corresponding increase of pressure. The design is such that the pressure rises to a value sufficiently above atmospheric to make possible the discharge of the mixture to the atmosphere.

Such a device is shown diagrammatically in its simplest form in Fig. 188. Steam at a high pressure enters the expanding steam nozzle designed to discharge it at a high velocity and with a pressure slightly lower than that carried in the condenser. The jet of steam passes through the space indicated as "entraining space" and into the "discharge tube." While passing through the entraining space, the jet picks up or entrains some of the noncondensible material held in that space and the mixture enters the discharge tube. The taper or flare of the discharge tube below the "neck" is so proportioned that the high velocity of the mixture is reduced and the pressure correspondingly increased to a value equal to or greater than atmospheric.

The entraining space is connected to that part of the

condenser in which the noncondensible gases collect. As this material is constantly removed from the entraining space by the steam jet, a continuous flow from condenser to air pump results.

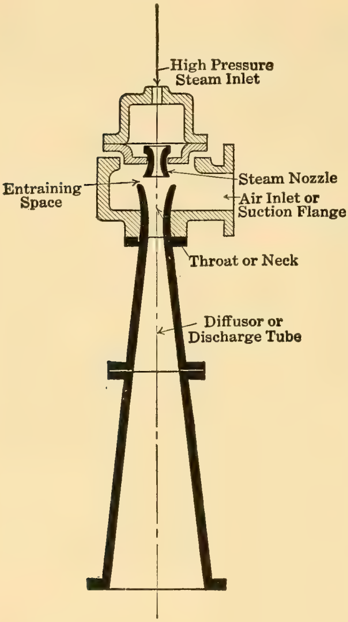


Fig. 188.—Steam-jet Type Vacuum Pump.

Experience has shown that a single stage pump like that illustrated in Fig. 188 is not the best possible arrangement. As a consequence, two and three stage pumps have been produced, the two stage being the type now generally used. Such a pump is illustrated in Fig. 189, and in Fig. 190 is shown one way in which it is connected into a condenser installation.

The device shown in Fig. 189 is called the Radojet Vacuum Pump, taking its name from the peculiar arrangement at the second stage which is described later. Instead of using one steam nozzle leading steam into the entraining space as shown in Fig. 188, a group of small nozzles is used. This is done for the purpose of breaking the steam up into a number of small jets, thus increasing the amount of air entrained by a given weight of steam.

The mixture discharged from the diffusor of the first stage enters the second entrainment space. Steam admitted at the second stage steam inlet expands through the peculiarly shaped nozzle indicated and is discharged at high velocity in the form of a circular sheet. This sheet entrains the mixture discharged from the first stage and carries it

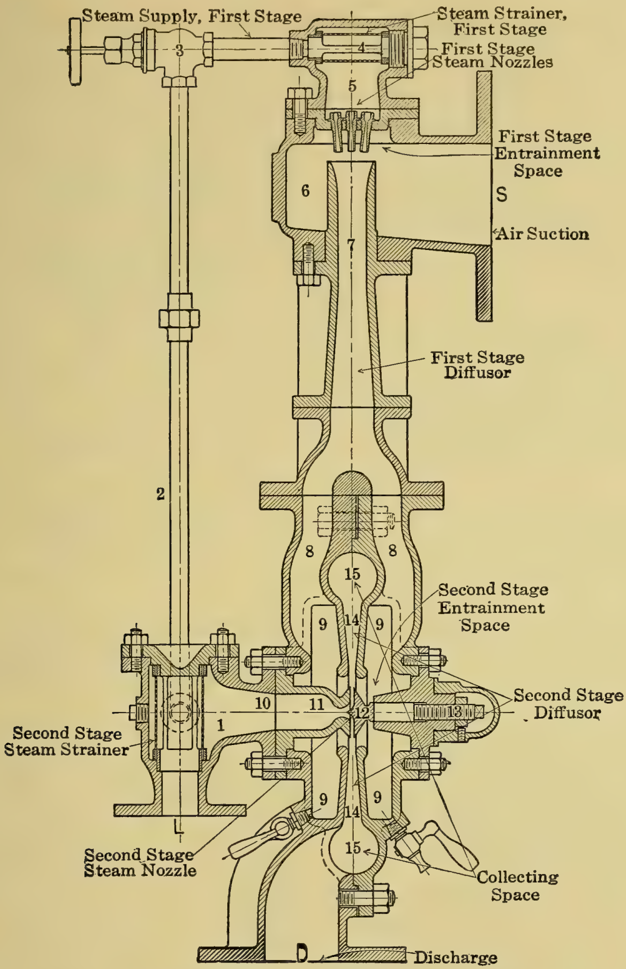


FIG. 189.—Cross-sectional View of the Radojet Vacuum Pump.

into the diffusor shown. This diffusor consists of two circular plates which are close together near the steam nozzle and separate gradually toward their circumferences. The mixture from the second diffusor is discharged into the

collecting space shown and flows out at the discharge flange.

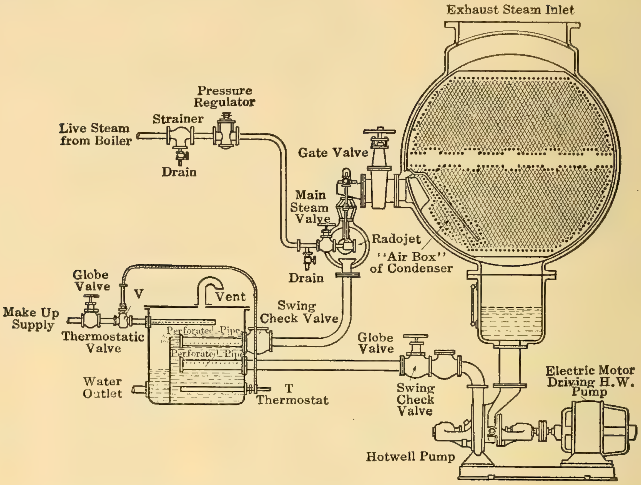


FIG. 190.—One Method of Connecting Radojet to Surface Condenser.

The radial flow from the second stage steam nozzle and through the second stage diffusor gives the device its trade name.

In the arrangement illustrated in Fig. 189, the second stage steam has to entrain and compress not only the " air " coming from the condenser but also all of the steam used in the first stage. This requirement leads to the use of large quantities of second stage steam. When thermal efficiency is of sufficient significance, it is customary to separate the first and second stages by interposing an " intercooler." This is merely a small surface condenser in which first stage steam and possibly vapors brought over from the condenser can be liquefied, leaving only saturated air to be handled by the second stage jet.

When such an intercooler is installed, some or all of the condensate from the main condenser is generally used as circulating water, so that the heat given up by steam and

air is caught in this condensate and returned to the boilers.

119. Water Required by Contact Condensers. The weight of circulating water required varies with the type of condenser and with the conditions of operation, such as initial temperature of water, vacuum desired, etc. It can be determined approximately by calculation and the values thus found must then be increased by such factors as experience has shown to be necessary.

In contact condensers the water and the condensate are discharged as a mixture and therefore have the same average discharge temperature.

Let t_1 = initial temperature of injection water in F.°;

t_2 = temperature at which mixture is discharged in F.°;

λ = total heat above 32° F. of steam as exhausted;

W = pounds of injection water per pound of exhaust steam.

Assuming the exhaust steam to be dry saturated, each pound of steam in condensing to water at a temperature of t_2 degrees must give up an amount of heat equal to λ minus the heat of the liquid at $t°_2$ or roughly $\lambda-(t_2-32)$ B.t.u. This same quantity must be absorbed by the injection water, while its temperature rises from t_1 to t_2 degrees. Each pound of water can then absorb approximately (t_2-t_1) B.t.u. and the pounds of injection water per pound of steam will be

$$W=\frac{\lambda-t_2+32}{t_2-t_1}. \quad . \quad . \quad . \quad . \quad . \quad (78)$$

The value of t_2 would be that corresponding to the absolute pressure in the condenser if it were not for the air and similar gases which exert some pressure. It is generally 10 or more degrees F. below the temperature corresponding to the vacuum. Values of t_2 in the neigh-

borhood of 110° to 125° F. are customary with reciprocating engines and values as low as 80° are used with high vacuums in connection with steam turbines.

The weight of water used per pound of steam as given by Eq. (78) will vary between about 15 for very low initial and moderate discharge temperature to about 50 with average initial and moderate discharge temperature. Experience shows that it is necessary to add 10 per cent or more to the values of W obtained from equation (78) to obtain the weight of water which will probably be used.

ILLUSTRATIVE PROBLEM

Find the quantity of water theoretically required per pound of steam condensed in a contact condenser in which a vacuum of 25.5 ins. of mercury is maintained when the barometer reads 29.5 ins. of mercury. The initial temperature of the water is 60° F.

The absolute pressure in the condenser is $29.5-25.5=4.0$ ins. of mercury and the steam tables give for this pressure, $\lambda=1115.0$ and $t_2=126$. Substituting in Eq. (78) gives

$$W=\frac{1115.0-126+32}{126-60}=15.5 \text{ approximately.}$$

120. Weight of Water Required by Non-contact Condensers. In the case of non-contact condensers there is no definite relation between the discharge temperature of the cooling water and that of the condensate. Experience shows that the discharge temperature of the circulating water is generally from 10 to 20 or more degrees lower than the temperature corresponding to the vacuum.

The temperature of the condensate (hot-well temperature) is often 15 or more degrees below that corresponding to the vacuum, but good design makes the hot-well temperature very closely approximate that corresponding to the vacuum.

Assuming

t_1 = initial temperature of injection water in F.°;
t_2 = final temperature of injection water in F.°;
t_c = temperature at which condensate is discharged, i.e., hot-well temperature, in F.°;
λ = total heat above 32° F. of steam as exhausted,

and

W = pounds of injection water per pound of exhaust steam.

The weight of water which must be circulated per pound of steam can be found as in the case of the contact condenser. It is given by

$$W=\frac{\lambda-t_c+32}{t_2-t_1}. \qquad (79)$$

Values in the neighborhood of 25 lbs. of water per pound of steam are common with low vacuums and 50 or more pounds are often used with vacuums over 28 ins. of mercury.

ILLUSTRATIVE PROBLEM

A surface condenser receives circulating water at a temperature of 65° F. and discharges it at a temperature of 80° F. It maintains a vacuum of 28.0 ins. with the barometer at 29.5, and the temperature of the condensate discharged to the hot well is equal to 85° F. Find the quantity of circulating water theoretically required.

This vacuum corresponds to an absolute pressure of $29.5-28.0=1.5$ ins. of mercury. Assuming this all due to steam (neglecting presence of air) the value of λ may be taken from the steam table as 1100.1 B.t.u. Substitution in Eq. (79) then gives

$$W=\frac{1100.1-85+32}{80-65}=69.9 \text{ approximately.}$$

121. Relative Advantages of Contact and Surface Condensers. The contact types are as a rule much cheaper

than the surface condensers, and they are less subject to serious depreciation, the tubes of surface condensers often corroding excessively in very short intervals of time. On the other hand, the injection of the cooling water into the condensing space in contact types results in the introduction of large quantities of dissolved gases, and much of this material is liberated under the reduced pressure, thus tending to increase the condenser pressure, that is, decrease the vacuum. Where pumps are used to carry away the mixture with contact condensers, these pumps have to handle a much larger quantity of water than the corresponding pump in a surface condenser, and the work of pumping this water out of the vacuum into the atmosphere combined with the additional work required of the pump which handles the "air" may partly balance the advantage of lower first cost of the contact type.

A surface condenser must always be installed where it is desirable to use the condensate as boiler feed, and it is generally used when very high vacuums (low absolute pressures) are to be maintained. The surface condenser is at a serious disadvantage, however, when required to handle the exhaust of reciprocating engines. The exhaust from such engines always contains large quantities of lubricating oil carried out of the cylinder, and unless this material is separated before the exhaust enters the condenser it is deposited on the outer surfaces of the tubes and decreases the conductivity of those surfaces. Such oil can be eliminated to a great extent before the exhaust enters the condenser by means of oil separators, which are generally made up of a series of baffles upon which the steam impinges and upon which the oil is caught.

122. Cooling Towers. The large quantity of circulating water required by condensing plants is often an item of great economic importance. When such plants are located near a river or near tide water, the circulating water can generally be procured for the cost of pumping.

When they are located in the middle of cities or in regions where water is scarce, the cost of water may be excessive or it may even be impossible to obtain a continuous supply equal to the demand of the condensers.

In such cases the condensing water is often circulated continuously, being cooled after each passage through the condensers. This cooling is generally done by exposure of a large surface to the air. The resultant evaporation of some of the water with the absorption of its latent heat of vaporization cools the remainder so that it can be used again. This sort of cooling may be effected by running the water into a shallow pond of large area, or by spraying it into the air over a small pond or reservoir or by passing it through a *cooling tower*.

Cooling towers are large wood or metal towers generally filled with some form of baffling devices. The hot water is introduced at the top and spread into thin sheets or divided up into drops as it descends. Air enters at the bottom and flows upward, cooling the water by contact and by the partial evaporation which results. The circulation of air may be natural, i.e., due to the difference of temperature between the air inside and out, in which case a stack is fitted to the top of the tower; or the circulation may be forced by fans located in the base of the tower. In the latter case the apparatus is called a *forced-draught cooling tower.*

CHAPTER XV

COMBUSTION

123. Definitions. Certain substances are known to chemists as **compounds,** because they can be separated by chemical processes into simpler substances. Thus water and many of the most familiar materials known to man are compounds which can be separated into two or more simpler materials.

Those substances which cannot be further broken up by the processes used in separating compounds are called **elements**; they are regarded as elemental, as the stones of which the compounds of nature are built up. About eighty-three of these elements are now known, but many of them are comparatively rare. Pure metals are all elements; the oxygen and nitrogen which are mixed to form the greater part of the atmosphere are elements; carbon, which forms the greater part of most fuels, is an element.

In many cases the *combination* of elements to form compounds is *accompanied by the liberation of heat,* and some of these combinations are used by the engineer for the purpose of obtaining heat in large quantities. When the elements which occur in fuels, such as coal, wood and petroleum, combine with oxygen, the process is spoken of as **combustion.** The quantity of heat liberated when a pound of any material combines with oxygen (burns) is called the **heat value** or **calorific value** of that material.

Fuels contain a great number of elements, but only three of these ordinarily take part in combustion and are therefore spoken of as combustibles. They are *carbon, hydrogen* and *sulphur.* The sulphur content is generally

very small, and the carbon and hydrogen are therefore the most important constituents.

The combustion of each of these elements will be considered in detail in the following sections, but before this can be done two other ideas must be developed.

The smallest particle of an element which can be conceived of as entering into combination to form a compound is known as an **atom** of that element. It has been found that the atoms of each element have an invariable and characteristic mass. The lightest atom is that of hydrogen, and its weight is considered unity. The atom of carbon is twelve times as heavy as that of hydrogen and carbon is therefore said to have an atomic weight equal to twelve. Similarly the atomic weight of nitrogen is fourteen and that of oxygen is sixteen.

The smallest particle which can be formed by the combination of atoms is known as a **molecule**. Like or unlike atoms may combine to form molecules. Thus two hydrogen atoms combine to form a molecule of hydrogen, and hydrogen gas as it ordinarily exists may be pictured as made up of a collection of such molecules. Similarly, gaseous oxygen and gaseous nitrogen may be pictured as collections of molecules which are made up of two like atoms.

When unlike atoms combine to form a molecule, they form a molecule of a compound. Obviously a molecule of any compound is the smallest particle of that compound which can exist.

For convenience, the different elements are represented by abbreviations; thus oxygen is represented by O, nitrogen by N, hydrogen by H, carbon by C and sulphur by S. When these abbreviations are written in chemical equations, such as will be given later, they stand for an atom of the substance. Hence O in a chemical equation would mean one atom of oxygen. The symbol O_2 is used to mean two atoms of oxygen in combination, hence, one molecule

of oxygen. The symbol $2O_2$ means two groups of two oxygen atoms in combination, hence two molecules of oxygen.

The simplicity and elegance of this system will become apparent as the chemical equations which follow are developed and explained.

124. Combustion of Carbon. Carbon can unite with oxygen or burn to form two different compounds—carbon monoxide (CO) and carbon dioxide (CO_2). The monoxide is formed by the combination of one atom of oxygen with one atom of carbon; the dioxide, by the combination of two atoms of oxygen with one of carbon. The dioxide, therefore, contains twice as much oxygen as does the monoxide.

Carbon burned to carbon monoxide has not combined with the largest possible quantity of oxygen, and combustion is therefore said to be *incomplete* in such cases. When, however, carbon dioxide is formed, the carbon has combined with as much oxygen as possible and combustion is said to be *complete*.

It will be shown later that much more heat is liberated when the dioxide is formed than when carbon burns to the monoxide. Hence, *when liberation of heat is the object* of combustion, the process should be so conducted as to result in the formation of the maximum quantity of dioxide and the minimum amount of monoxide.

125. Combustion to CO. The combustion of carbon and oxygen to form the monoxide can be represented by the equation

$$C+O=CO, \qquad (80)$$

or by the equation

$$2C+O_2=2CO. \qquad (81)$$

The former is the simpler and will be considered first, but the latter is the more perfect and indicates more to the trained eye than does the simpler form.

The simple equation states that one atom of carbon combined with one atom of oxygen to form one molecule

of carbon monoxide. It can, however, be so interpreted as to show much more than this. *The carbon atom is twelve times as heavy as the hydrogen atom, while the oxygen atom is sixteen times as heavy as that of hydrogen.* The equation

$$C+O=CO,$$

therefore, shows that an atom, which is twelve times heavier than the hydrogen atom, unites with one which is sixteen times heavier than the hydrogen atom to form a molecule which is $28(=12+16)$ times heavier than the hydrogen atom.

In other words, the weights of carbon and oxygen combining are in the ratio of $\frac{12}{16}=\frac{3}{4}=\frac{1}{1\frac{1}{3}}$. If a sufficient number of carbon atoms to weigh one pound be used, a quantity of oxygen weighing $1\frac{1}{3}$ lbs. will be required to combine with them to form carbon monoxide. The resultant carbon monoxide will contain the pound of carbon and the $1\frac{1}{3}$ lbs. of oxygen and will therefore weigh $2\frac{1}{3}$ lbs.

The same weight relations would hold irrespective of the weight of carbon used, and the simpler equation may therefore be put

$$1 \text{ weight of } C+1\tfrac{1}{3} \text{ weights of } O=2\tfrac{1}{3} \text{ weights of } CO. \quad (82)$$

ILLUSTRATIVE PROBLEM

To illustrate the use of this equation, assume that 9 lbs. of carbon are burned to carbon monoxide and that it is desired to find the weight of oxygen used, and the weight of the product. The weight of oxygen used must be $1\frac{1}{3}$ times the weight of carbon, that is, $1\frac{1}{3}\times 9=12$ lbs. The weight of the product must be $2\frac{1}{3}$ times the weight of the carbon, that is $2\frac{1}{3}\times 9=21$ lbs.; or, it must be the weight of the carbon burned plus the weight of the oxygen used, that is, $9+12=21$ lbs.

In general, the oxygen used for combustion is obtained from the *atmosphere*, which may be considered as a *mechan-*

ical mixture of oxygen and nitrogen in unvarying porportions. These proportions are roughly, 0.23 of oxygen to 0.77 of nitrogen by weight, or 0.21 of oxygen to 0.79 of nitrogen by volume, as shown in Table VIII. The weight of air which contains one pound of oxygen is therefore $\frac{0.23+0.77}{0.23}=4.35$ lbs., and this weight of air contains $4.35-1=3.35$ lbs. of nitrogen.

In the problem previously considered it was found that 12 lbs. of oxygen would be required to burn 9 lbs. of carbon to CO. The total weight of air required to obtain this oxygen will be $12\times 4.35=52.2$ lbs. and it will contain $52.2-12=40.2$ lbs. of nitrogen.

By simple arithmetical calculations of the type just given all the weight relations involved in the combustion of C to CO can be determined. The volume of air required in any given case can be found by multiplying the weight of air by the specific volume as given in Table VIII. Thus, in the illustrative problem already considered, it was found that 52.2 lbs. of air would be required to burn 9 lbs. of C to CO. The volume of this air at 62° F. would be $52.2\times 13.14=685.9$ cu.ft.

It is found that a quantity of heat equal to about 4500 B.t.u. is liberated per pound of carbon burned to CO; that is the *calorific value of* C *burned to* CO *is* 4500 *B.t.u.*

Returning now to Eq. (81), which was said to be more useful than the simpler form given as Eq. (80), it will be necessary to consider a rather simple law of gases. It has been shown experimentally that *equal volumes of all gases contain the same number of molecules when at the same temperature and pressure.* This statement is known as **Avogadro's Law.**. It has also been shown that the molecules of gaseous oxygen contain two atoms.

The equation in question,

$$2C+O_2=2CO$$

can therefore be read, two atoms of carbon combine with *one molecule of oxygen* to form *two molecules of carbon monoxide.* But, if every molecule of O yields two molecules of CO it follows from Avagadro's law that the CO *formed will occupy twice the volume of the oxygen used* if measured at the same temperature and pressure. If the equation be imagined as containing a numeral 1 before the O_2, it will be obvious that the coefficients of the terms representing gas molcules give volume relations directly. This equation therefore gives both volume and weight relations.

TABLE VIII

PROPERTIES OF AIR

Considering it to consist only of nitrogen and oxygen.

	Relative Proportions.		Ratio of N to O.		Ratio of Air to O.	
	Exact.	Approx.	Exact.	Approx.	Exact.	Approx.
By weight..	0.766 N 0.234 O	0.77 N 0.23 O	3.27	3.35	4.27	4.35
By volume..	0.791 N 0.209 O	0.79 N 0.21 O	3.78	3.76	4.76	4.76

Spec. wt. at Atmos. Press. (Lbs. per Cu.ft.)		Spec. Vol. at Atmos. Press. (Cu.ft. per Lb.)	
At 32° F.	At 62° F.	At 32° F.	At 62° F.
0.08072	0.07609	12.39	13.14

Weight of air containing one pound of oxygen is approximately 4.35 lbs.

126. Combustion to CO_2. The combination of carbon and oxygen to form the dioxide is represented by the equation

$$C+O_2=CO_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (83)$$

which shows that one atom of carbon (twelve times heavier than hydrogen) combines with two atoms of oxygen (each sixteen times heavier than hydrogen) to form a molecule of CO_2, which is forty-four times heavier than an atom of hydrogen. Therefore the weight of carbon and oxygen combining are as $\frac{12}{2\times16}=\frac{3}{8}=\frac{1}{2\frac{2}{3}}$, so that $2\frac{2}{3}$ lbs. of oxygen are required to burn a pound of carbon to carbon dioxide. Writing this in the form of an equation, gives

$$1 \text{ weight of } C+2\tfrac{2}{3} \text{ weights of } O=3\tfrac{2}{3} \text{ weights of } CO_2. \quad . \quad (84)$$

The weight of air required can readily be found by multiplying the required oxygen by the number 4.35, previously shown to be the number of pounds of air containing one pound of oxygen. Thus, the required air is $2\frac{2}{3}\times4.35=11.57$ pounds per pound of C burned to CO_2. This number is commonly rounded out to 12 in engineering literature.

The equation given shows volume relations directly. It is evident, therefore, that one molecule of O yields one molecule of CO_2, and hence that the volume of the product is exactly equal to the volume of the oxygen used in forming it if measured at the same temperature and pressure. This is a very important relation, and is often made use of in engineering calculations.

Experiment shows that when carbon burns to the dioxide about 14,600 B.t.u. are liberated per pound of carbon burned, that is, the *calorific value of* C *burned to* CO_2 in 14,600.

127. Combustion of CO to CO_2. Since carbon which has burned to carbon monoxide has not combined with the greatest possible quantity of oxygen, the monoxide can take up more oxygen by burning to the dioxide. This process is represented by the formula

$$2CO+O_2=2CO_2, \quad . \quad . \quad . \quad . \quad . \quad (85)$$

which shows that two molecules of monoxide combine with one molecule of oxygen to form two molecules of the dioxide. The volume of CO_2 formed is therefore equal to that of the CO burned.

So far as the ultimate result is concerned, it makes no difference whether carbon is burned directly to CO_2 or is first burned to CO and then the CO is burned to CO_2. The total oxygen used per pound of carbon burned to CO_2 and the total heat liberated per pound of carbon burned to CO_2 are the same in both cases.

Thus, for the oxygen, one pound of C burned to CO_2 requires $2\frac{2}{3}$ lbs. of oxygen; but one pound of C burned to CO requires $1\frac{1}{3}$ lbs. of oxygen, and $1\frac{1}{3}$ lbs. more will be required when this CO is burned to CO_2. The result is therefore the same.

For heat liberated, one pound of C burned to CO_2 liberates about 14,600 B.t.u.; but one pound of C burned to CO liberates about 4500 B.t.u. and 10,100 B.t.u. are liberated when this CO is burned to CO_2. Since the sum of 4500 and 10,100 is equal to 14,600 the result is again the same.

Data on the combustion of C to CO and CO_2 and the combustion of CO to CO_2 are collected in convenient form in Table IX.

128. Conditions Determining Formation of CO and CO_2. Excluding certain complicated considerations which are not of great importance in steam-power engineering, it may be said that when carbon is being burned at a certain rate (pounds per unit of time) the amount of oxygen brought into contact with the carbon determines whether the carbon burns to CO or to CO_2. If enough or more than enough oxygen to burn the carbon to CO_2 is brought into contact, that oxide will be formed. If there is not enough to burn all the carbon to the dioxide, both oxides are formed in certain proportions, which can be calculated.

Since combustion to CO yields only 4500 B.t.u. per

pound of C and combustion to CO_2 yields 14,600 B.t.u. per pound of C, the importance of supplying sufficient oxygen to burn all carbon to the dioxide in cases where the liberation of the maximum quantity of heat is desirable is obvious. In actual practice the oxygen is furnished by supplying air and it is found necessary in most cases to supply more than the amount of air theoretically required in order to insure burning all, or even nearly all, of the carbon to the dioxide. This comes from the great *difficulty met in obtaining contact between the oxygen of the air and the carbon* which is to be burned, that is, in bringing all the oxygen of the air into intimate contact with the fuel being burned in real apparatus.

TABLE IX

COMBUSTION DATA FOR CARBON

(Per pound of carbon.)

Product.	Oxygen Required.		Nitrogen Accompanying Oxygen.	
	Pounds.	Cu.ft. at 62° F. and 14.7 Lbs.	Pounds.	Cu.ft. at 62° F. and 14.7 Lbs.
CO	1.333	16.0	4.46	60.1
CO_2 from C	2.667	32.0	8.92	120.2
CO_2 from CO	1.333	16.0	4.46	60.1

Product.	Air Required.		Quantity of Product (N not included).		Heat Liberated.
	Pounds.	Cu.ft. at 62° F. and 14.7 Lbs.	Pounds.	Cu.ft at 62° F. and 14.7 Lbs.	
CO	5.79	76.1	2.33	32.0	4,500
CO_2 from C	11.58	152.2	3.67	32.0	14,600
CO_2 from CO	5.79	76.1	3.67	32.0	10,100 per lb. of C in CO 4,300 per lb. of CO

The air in excess of that theoretically required to burn all the carbon completely is spoken of as **excess air.** In the form of an equation, this statement is equivalent to

$$\text{Air supplied} - \text{air theoretically required} = \text{excess air.} \quad (86)$$

It is customary to express the quantity of excess air in terms of a numerical factor known as the **excess coefficient.** This coefficient is defined as the *number by which the quantity of air theoretically required must be multiplied to give the quantity of air actually used.* In the form of an equation this gives

$$\text{Excess coefficient} \times \text{air theoretically required} = \text{air actually used.} \quad (87)$$

ILLUSTRATIVE PROBLEM

Taking data from the illustrative problem previously considered, assume that 9 lbs. of carbon are burned in air to CO_2. Each pound theoretically requires 11.57 lbs. of air, so that the theoretical air-supply for this case would be $9 \times 11.57 = 104.13$ lbs. If in a real case 150 lbs. of air are supplied, the excess coefficient is equal to $150 \div 104.13 = 1.44$.

129. Flue Gases from Combustion of Carbon. The gases resulting from the combustion of fuels are known in engineering as the *products of combustion or flue gases,* because they are the gases passing through the flues or passages leading from furnaces in which fuel is burned and to the stacks which serve to carry off the gases.

It has already been shown that the CO_2 formed by the combustion of carbon has the same volume as the oxygen which is used in forming it. Therefore, if the air supplied in a given case just equaled that theoretically required for combustion to CO_2 and if all of the oxygen were used, the CO_2 formed would *merely replace the oxygen* in the air. The theoretical proportions of the flue gas would then be 0.21 of CO_2 and 0.79 of N by volume.

If real flue gases obtained by burning carbon in air are found to contain less than 21 per cent of CO_2, the combustion has evidently not yielded theoretically perfect flue gases. The trouble may be due to an excess or to a deficiency of air. If there is an excess of air there will be oxygen present in the flue gases; if there is a deficiency there will be CO present in the flue gases. An analysis of these gases for oxygen and for CO would therefore indicate the source of trouble and the remedy to be provided.

The curve to the right of the central vertical line in Fig. 191 shows the theoretical decrease in volume per

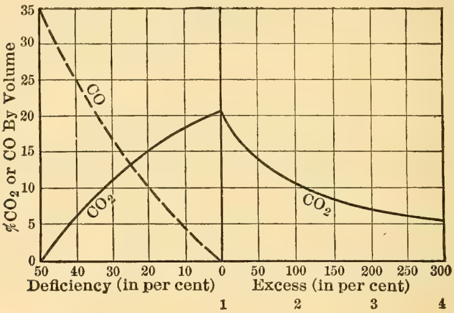


FIG. 191.—Effect of Air Supply on Flue Gas Analysis.

cent of CO_2 in flue gases as the excess air increases. The single numbers 1, 2, 3 and 4 indicate the excess coefficients corresponding to the various percentages of excess air.

The curves to the left give the theoretical decrease in volume per cent of CO_2 and the theoretical increase in volume per cent of CO as the air supplied is decreased below that theoretically required for complete combustion.

130. Combustion of Hydrogen. Hydrogen combines with oxygen, or burns, to form water. The equation for this reaction is

$$2H_2+O_2=2H_2O, \quad \ldots \ldots \quad (88)$$

which indicates that two molecules of hydrogen combine with one molecule of oxygen to form two molecules of water. In terms of volumes, two volumes of hydrogen combine with one of oxygen to form two of *gaseous* water, that is, water in the form of highly superheated vapor. As the water is cooled down it will obviously approach and finally reach the liquid condition, with a rapid decrease in volume quite different from that experienced by a gas under similar conditions, so that the volume relations hold only at high temperatures.

The weight relations can be calculated as in other cases, starting from the fact that four weights of hydrogen combine with thirty-two weights of oxygen to form 36 weights of water. The weights of hydrogen and oxygen are therefore in the relation of $\frac{4}{32}=\frac{1}{8}$.

The heat liberated when one pound of hydrogen burns to water is equal to about 62,000 B.t.u. This is the quantity of heat which could be obtained if one pound of hydrogen at, say, room temperature, and mixed with the requisite quantity of oxygen, were ignited and the resultant water were then cooled down to the initial temperature. *During the cooling of the water it would partly or entirely condense and thus give up some or all of its latent heat of vaporization.* This heat would obviously be included in the calorific value just given.

In many pieces of engineering apparatus in which hydrogen is burned the products of combustion are not cooled to such an extent that the water is condensed. The latent heat of vaporization would not be liberated under such conditions, but would remain bound up with the water vapor. *When the water is not condensed* the heat liberated is only about 52,000 B.t.u. per pound of hydrogen. This number is known as the *lower calorific value* of hydrogen, while 62,000 is known as the *higher calorific value.*

Data on the combustion of hydrogen are given in Table X.

TABLE X

COMBUSTION DATA FOR HYDROGEN

(Per pound of hydrogen)

Product.	Oxygen Required.		Nitrogen Accompanying Oxygen.	
	Pounds.	Cu.ft at 62° F. and 14.7 Lbs.	Pounds.	Cu.ft. at 62° F and 14.7 Lbs.
H_2O	8	96	26.8	361

Product.	Air Required.		Quantity of Product (N not included).		Heat Liberated.
	Pounds.	Cu.ft. at 62° F. and 14.7 Lbs.	Pounds.	Cu.ft. at 62° F. and 14.7 Lbs.	
H_2O	34.8	457	9	Liquid 0.144	62,000 52,000

131. Combustion of Hydrocarbons. Many of the fuels used by the engineer contain *compounds of hydrogen and carbon* which are called **hydrocarbons**. One of the best examples is *methane* (CH_4), which forms the greater part of all the so-called natural gas.

All of these hydrocarbons burn to CO_2 and H_2O if the supply of oxygen is great enough. If there is a deficiency of oxygen, combustion is incomplete and generally results in the formation of CO_2, H_2O, CO, C in the form of soot, and other products which need not be considered here.

For complete combustion the requisite oxygen and air can be determined as in previous cases by means of chemical equations. Thus for methane the equation is

$$CH_4+2O_2=CO_2+2H_2O, \quad . \quad . \quad . \quad (89)$$

which shows that sixteen (12+4) weights of methane combine with sixty-four (2×2×16) weights of oxygen to form forty-four (12+32) weights of carbon dioxide and thirty-six (4+32) weights of water.

The calorific value of hydrocarbons is generally assumed to be equal to the sum of the heat values of the carbon and hydrogen contained in one pound of the material. Thus, if C represent the fraction of a pound of carbon contained in one pound of the hydrocarbon and if H represent the fraction of a pound of hydrogen contained therein, the common assumption would make the higher calorific value of the hydrocarbon

$$(C\times 14{,}600)+(H\times 62{,}000)\text{ B.t.u.} \quad . \quad . \quad (90)$$

The results obtained in this way do not generally check well with the experimentally determined values, and it is best to use the latter when they are available.

132. Combustion of Sulphur. Sulphur forms several different oxides, but when burned under engineering conditions it is generally assumed to form only the dioxide SO_2. The chemical equation for such combustion is

$$S+O_2=SO_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (91)$$

and since the atomic weight of sulphur is 32, this equation shows that equal weights of sulphur and oxygen combine to form the dioxide.

The combustion of sulphur to SO_2 liberates about 4000 B.t.u. per pound of sulphur.

133. Combustion of Mixtures. It is often necessary to obtain approximate calorific values of combustible materials which, without great error, can be considered as mixtures of combustible and non-combustible elements. If there is oxygen present in the mixture it is assumed to be combined with hydrogen in the form of water, so that the *uncombined* or *available hydrogen* per pound of material is given by the expression

$$\text{Available H}=H-\frac{O}{8}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

in which H and O respectively represent the fractions of a pound of hydrogen and oxygen in one pound of material.

The calorific values of such a mixture containing carbon, hydrogen and sulphur would then be given approximately by the equation

$$\text{Higher B.t.u.} = 14{,}600\text{C} + 62{,}000\left(\text{H} - \frac{\text{O}}{8}\right) + 4000\text{S}, \quad (93)$$

in which the letters stand respectively for the fractions of a pound of each of the elements present in one pound of the mixture. Similarly the lower calorific value would be (approximately)

$$\text{Lower B.t.u.} = 14{,}600\text{C} + 52{,}000\left(\text{H} - \frac{\text{O}}{8}\right) + 4000\text{S}, \quad (94)$$

and the oxygen required will be

$$\text{Pounds of O} = 2\tfrac{2}{3}\text{C} + 8\left(\text{H} - \frac{\text{O}}{8}\right) + \text{S}. \quad . \quad . \quad (95)$$

134. Composition of Flue Gases. It was shown in Section 129 that the flue gases resulting from combustion of carbon to carbon dioxide with the theoretical amount of air would consist of 21 per cent carbon dioxide and 79 per cent nitrogen by volume. It was also shown that variation from this composition in any real case can be interpreted to show the cause of such variation.

It is necessary to note that the figures given apply only to the case of pure carbon. If, for instance, hydrogen is burned with air, no carbon dioxide can result, since no carbon is present and the composition of the flue gases must therefore be quite different from what was indicated above.

As a matter of fact, the flue gases resulting from the combustion of hydrogen with the theoretical air supply would, if cooled down to ordinary temperatures, consist of nitrogen saturated with water vapor. All water vapor in excess

of that required to fill the space occupied by the nitrogen (at the existing temperature) would condense out as liquid water.

All real fuels contain carbon and hydrogen and often sulphur as well. The theoretical composition of flue gases obtainable with real fuels is therefore quite different from that indicated for pure carbon. The theoretical air supply for such fuels contains just enough oxygen to burn the combustible constituents and the quantity of nitrogen which must accompany that oxygen. Irrespective of total quantities involved, the oxygen must represent about 21 per cent of the volume of the air and the nitrogen 79 per cent.

After combustion is completed, some of this oxygen has been converted into carbon dioxide occupying the same volume as the oxygen from which it was made (when reduced to the same pressure and temperature). Obviously with real fuels the carbon dioxide must occupy less volume than all of the oxygen and therefore must form less than 21 per cent of the volume of the final products of combustion.

The figures given below will serve to indicate the variations which occur with real fuels. In each case a typical analysis has been assumed for the fuel named and it has beeen assumed that it is burned with the theoretical air supply. It should be noted that the percentage of carbon dioxide given holds only for the particular analysis of fuel assumed.

Fuel.	Per Cent Carbon Dioxide in Flue Gases.
Bituminous Coal	18.4
Wood	20.1
Petroleum Oil	15.4
Natural Gas	11.7
Blast Furnace Gas	25.1

The high value for blast furnace gas is explained by the fact that this gas contains a large amount of carbon dioxide.

so that this quantity appears in the flue gas in addition to that formed by combustion of other constituents of the gas.

135. Temperature of Combustion. If combustion of any material could be carried on inside an ideal vessel which did not absorb nor transmit heat, the heat liberated during the combustion could not escape from the space within the vessel.

If the vessel contained initially only the combustible and the oxygen or air required to burn it, the products of combustion would be the only material contained within the vessel after the completion of combustion. Under such circumstances the heat would be used in raising the temperature of the products of combustion, and the process could be pictured as though all of the combustion occurred first, forming the products of combustion without change of temperature, and then the liberated heat raised the temperature of these products.

Knowing the weight of each of these products and the quantity of heat required to raise the temperature of one pound of each of them one degree, the amount of heat required to raise all of them one degree could be found by multiplying the two known values. Thus, if carbon had been burned in oxygen to CO_2 with the theoretical oxygen supply, the vessel would contain only carbon dioxide. To raise the temperature of one pound of carbon dioxide one degree requires an amount of heat equal to the specific heat of that gas. Therefore, if W represents the weight of CO_2 formed and C represents its specific heat, the amount of heat required to raise the temperature of all of the CO_2 one degree would be $W \cdot C$ B.t.u. If Q B.t.u. were liberated by the combustion, the *temperature rise* in degrees would therefore be given by

$$\text{Temp. rise} = \frac{Q}{WC}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (96)$$

and if the initial temperature had been t_0 degrees, the *final temperature* would be

$$t = t_0 + \frac{Q}{WC}. \quad \ldots \ldots \ldots \quad (97)$$

A final temperature figured in this way is called the **theoretical temperature of combustion.** It can never be attained in practice because of heat lost to surroundings and because of other losses which need not be considered here.

Theoretical temperatures of combustion are, moreover, nearly always calculated on the assumption that the *specific heats of gases* are constants, whereas they really *increase with the temperature.* It therefore follows that temperatures determined on the assumption of constant specific heat will be too high for this reason also.

When gases are heated there are two distinctly different limiting possibilities; the volume occupied by the gases may remain constant or the pressure exerted by the gases may remain constant while the volume increases. In the case of constant volume all the heat added to the gases must be used for raising the temperature; the amount of heat required per pound per degree under these conditions is known as the specific heat at constant volume and is designated by C_v.

When, however, the volume is allowed to increase at such a rate as to keep the pressure constant the heat supplied must not only raise the temperature, but must also do whatever external work is done in displacing (pushing out of the way) surrounding mediums. The heat required per pound per degree under these conditions is known as the specific heat at constant pressure and is represented by C_p. It is always greater than C_v by the amount of heat required to do the external work accompanying a rise of temperature of one degree.

Thus, in the case assumed above, had the vessel been so constructed that its internal volume did not change,

the specific heat at constant volume would be used. On the other hand, had the vessel been fitted with a movable piston arranged to move outward at such a rate as to maintain constant pressure within the vessel as the temperature rose, the specific heat at constant pressure would be used.

In most cases the material burned is not pure carbon, but a fuel containing carbon, hydrogen and sulphur, and as air is generally used to furnish the oxygen, the products of combustion will contain not only the oxides of carbon, hydrogen and sulphur, but inert nitrogen as well. The temperature rise is determined in the same way, however, by dividing the heat liberated by the amount required to raise the temperature of the products one degree. Thus if $W_1, W_2, W_3 \ldots W_n$ stand for the weights of the various products and $C_1, C_2, C_3 \ldots C_n$ for their respective specific heats, the *theoretical temperature rise* is given by

$$\text{Temp. rise} = \frac{Q}{W_1C_1 + W_2C_2 + W_3C_3 + \ldots + W_nC_n}, \qquad (98)$$

and the *theoretical temperature of combustion* is given by

$$t = t_0 + \frac{Q}{W_1C_1 + W_2C_2 + W_3C_3 \ldots W_nC_n}, \qquad (99)$$

if t_0 stands for the initial temperature.

PROBLEMS

1. Assume 10 lbs. of C burned to CO. Determine the quantity of oxygen required, the quantity of air required, the quantity of nitrogen in this air, and the quantity of heat liberated.

2. What will be the volume of the CO formed as above if measured at 62° F. and 14.7 lbs. pressure?

3. Assume 15 lbs. of C burned to CO_2. Determine the quantities of oxygen and air required, the quantity of nitrogen contained in this air, and the quantity of heat liberated.

4. What will be the volume of the CO_2 formed from 15 lbs. of carbon if measured at 62° F. and 14.7 lbs.?

5. What will be the volume of the flue gases formed by the combustion of 11 lbs. of carbon to CO_2 with the theoretical air supply?

6. The quantity of CO obtained by the combustion of 8 lbs. of carbon is burned to CO_2 with the theoretical amount of oxygen. Determine the quantities of oxygen and air required, the amount of nitrogen contained in this air, and the quantity of heat liberated.

7. Assume 5 lbs. of C burned in air to CO_2 with an excess coefficient of 1.5. Determine the quantities of oxygen and air supplied, the heat liberated and the composition of the flue gases.

8. The composition of flue gases resulting from the combustion of carbon in air is found to be 21% of CO_2 and 79% of N by volume. What is the value of the excess coefficient?

9. An analysis of flue gases resulting from the combustion of carbon in air shows 12% of CO_2 by volume and no CO. The gases are not analyzed for O or N. What can you say with regard to the air supply?

10. Three pounds of hydrogen burn with theoretical oxygen supply. Determine the weight of oxygen and air used, the weight of the resultant water and the weight of the flue gas.

11. Determine the heat liberated in the preceding problem if the water vapor is condensed and if it is not condensed.

12. How much hydrogen would have to be burned to obtain 20 lbs. of water?

13. The chemical formula of methane is CH_4. If one pound of methane is burned with theoretical air supply, what weight of air will be used, and what will be the weight of the flue gases?

14. What would be the percentage composition of the flue gases of the preceding problem on a weight basis?

15. The chemical formula of ethane is C_2H_6. Determine the calorific value of this material by means of the formula given in the text.

16. A certain material is found to have the following analysis on a weight basis: C, 85%; H, 12%; O, 1%; S, 2%. Determine the calorific value of this material by means of the formula given in the text, assuming that all the oxygen present is in combination with hydrogen.

17. Determine the amount of oxygen required to completely burn 3 lbs. of the material described in the preceding problem.

18. One pound of carbon is burned to CO_2 in pure oxygen in a vessel so arranged as to maintain constant internal pressure. The specific heat of CO_2 at constant pressure and at ordinary temperatures is about 0.21. Calculate the theoretical temperature

rise and the temperature of combustion, using this value of the specific heat and assuming an initial temperature of 60° F.

19. Make the same calculations as called for in the preceding problem, but using the value 0.27 for the specific heat of CO_2. This is more nearly the average value of the specific heat over the range of temperature existing in such a case.

20. The hydrocarbon ethylene is represented by the chemical formula C_2H_4. Assume that one pound of this material is burned in air within a vessel arranged to maintain the products at constant pressure and that the excess coefficient is 1.5. Determine the theoretical temperature of combustion if the initial temperature is 60° F., the mean specific heat of CO_2 is 0.27, that of H_2O is .61, that of N is 0.27, and that of O is 0.24.

CHAPTER XVI

FUELS

136. Commercial Fuels. In engineering practice anything which is combustible and which can be procured in large quantities *at a reasonable cost* is called a fuel. The principal commercial fuels are:

a. Solid
- (1) Coal.
- (2) Wood and wood wastes.
- (3) Vegetable wastes.

b. Liquid
- (1) Crude petroleum or natural oil.
- (2) Various products made from petroleum.
- (3) Methyl and ethyl alcohol.

c. Gaseous
- (1) Natural gas.
- (2) Artificial or manufactured gases.

Coal is by far *the most extensively used fuel* because of its abundance and relative cheapness in most localities. However, in oil-producing regions the *crude oil* and some of the products made from it are more often the commonly used fuel, particularly if good coal is not mined in the immediate vicinity.

Wood is, in general, too valuable to be used exclusively as a fuel excepting on the frontiers where wooded territory is being opened up and where coal cannot be procured excepting at prohibitively high cost. Wood wastes, on the other hand, are very often used for fuel in the industries producing them.

Vegetable wastes, like wood wastes, are essentially of local value, being practically entirely consumed by the industries producing them.

Natural gas is in many respects an ideal fuel, and is extensively used for power production in some localities. The diminution in the quantity available, the consequent rise in the price, the great economy achieved in burning this gas in gas engines and the increased use of the gas for domestic purposes are, however, gradually eliminating this fuel from the steam-power field.

Artificial gases have never been extensively used for the generation of steam, as it is generally cheaper to burn the materials from which the gases are made, rather than to convert them into gas and then to burn the gas under boilers. This condition may change in the future when better markets have been opened up for some of the by-products which can be obtained from artificial gas plants.

137. Coal. The word coal is used as the name of a great group of natural fuels which consist of more or less *metamorphosed vegetable remains.* At one end of the group is the material known as *peat,* which is only slightly changed from the original vegetable substance; at the other end is the *graphitic anthracite* which has undergone such radical metamorphosis that practically all of the original vegetable material excepting carbon and ash has been eliminated.

A common, rough classification of the coals in the order of age, or of completeness of carbonization is,

1. Peat or turf.
2. Lignite (brown or black).
3. Sub-bituminous coal.
4. Bituminous coal.
5. Semi-bituminous coal.
6. Semi-anthracite.
7. Anthracite.
8. Graphitic anthracite.

The divisions are not at all exact, as they depend partly upon chemical composition and partly upon physical properties.

Another classification of a more exact variety is that given in Table XI and partly illustrated in Fig. 192, which gives what is known as Mahler's curve. It is for United States coals only. The terms used in this classification are explained in subsequent paragraphs.

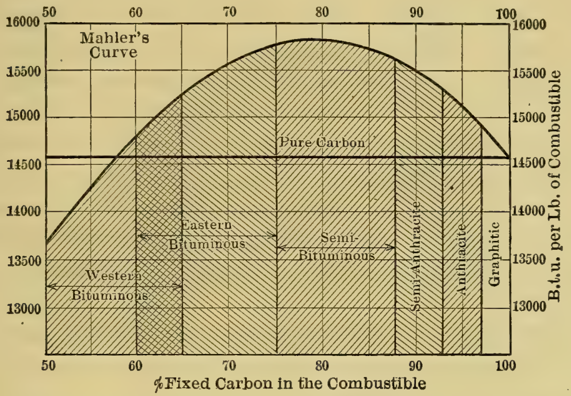


FIG. 192.—Mahler's Curve for United States Coals.

TABLE XI

CLASSIFICATION OF COALS

Division.	Per cent of Fixed Carbon in Combustible.	Per cent of Volatile Matter in Combustible.	Calorific Value, B.t.u. per Pound of Combustible.
Graphitic	100 to 97	0 to 3	14,600 to 14,900
Anthracite	97 to 92.5	3 to 7.5	14,900 to 15,300
Semi-anthracite	92.5 to 87.5	7.5 to 12.5	15,300 to 15,600
Semi-bituminous	87.5 to 75	12.5 to 25	15,600 to 15,900
Eastern bituminous	75 to 60	25 to 40	15,800 to 14,800
Western bituminous	65 to 50	35 to 50	15,200 to 13,700
Lignite	under 50	over 50	13,700 to 11,000

The *graphitic anthracite* occurs in very small quantities and mostly in Rhode Island. With a few minor exceptions the *anthracites* occur only in Eastern Pennsylvania and the

semi-anthracites are almost entirely confined to the western edge of this field.

The *semi-bituminous* coals are found on parts of the eastern border of what is known as the Appalachian coal field, extending from central Pennsylvania through the intermediate States to the northern part of Alabama. The greater part of this enormous bed consists of *eastern bituminous* coal. *Western bituminous* coals are found in large beds in the central part of the United States, principally in the States of Illinois, Indiana and Kentucky on the east of the Mississippi River, and in Iowa, Kansas and Texas to the west of that river.

Lignite is found in small quantities in nearly all of the western half of the United States and in large beds in the Dakotas, Texas, Arkansas, Louisiana, Mississippi and Alabama.

Peat is distributed in small beds throughout practically all of the United States and is continually forming in many marshes and on low-lying lands.

138. Coal Analyses. Two different coal analyses are in use, the simpler being known as the **proximate analysis** and the more exhaustive being called the **ultimate analysis**. Both are made and reported on a weight basis.

The proximate analysis assumes coal to contain four different and separable things, which are called *fixed carbon*, volatile hydrocarbon or *volatile matter* or volatile, *moisture* and *ash*.

Moisture is determined by maintaining a small quantity of finely ground coal at a temperature of about 220° F. for one hour. The material lost during this time is assumed to be moisture only and is reported as such. Coal from which the moisture has been driven in this way is called **dry coal**.

Volatile matter is determined by heating a sample from which the moisture has been driven, or a fresh sample. The coal is maintained at a red to white heat with exclu-

sion of air until there is no further loss of weight. In the case of a previously dried sample the loss under these conditions is called volatile hydrocarbon. If the sample was not previously dried a separate moisture determination is made on a similar sample and the weight of volatile is found by difference.

Fixed carbon is found by combustion of a sample from which the moisture and volatile have been driven, the loss under these conditions being assumed to be entirely due to the combustion of carbon.

Ash is the name given to the incombustible material left behind after determining the fixed carbon.

The volatile hydrocarbons and the fixed carbon as determined in the proximate analysis are assumed to be the only combustible parts of the coal and their sum is called *the combustible.*

Proximate analyses are reported in three different ways: On *coal as received,* on *dry coal,* and on *combustible.*

Since the water content of a sample of coal received at any plant is largely a matter of the weather conditions during shipment, the best idea of the character of a coal can be obtained by excluding the consideration of its moisture content. It is generally best, therefore, to convert analyses to a dry coal basis, that is, recalculate the percentages of volatile, fixed carbon and ash on the assumption that the analysis was made on the weight of coal which would result from drying the sample that was actually used. Excessive moisture is, however, undesirable for steam-raising purposes, and the amount of moisture should therefore be determined in every case.

Ash is also more or less a matter of accident in that the amount contained is largely determined by the care used in mining and subsequent cleaning of the coal. While it has a very appreciable effect upon the character of the material as a fuel it really has little connection with the combustible part of the fuel. For purposes of classifica-

tion, therefore, the ash should also be eliminated and the analysis given on the basis of combustible.

Sulphur is sometimes reported with a proximate analysis. In making such an analysis the greater part of the sulphur is really driven off with, and regarded as, part of the volatile, so that when the sulphur content is desired it must be determined by a separate analysis.

The *ultimate analysis* attempts to separate the dry combustible into the various elements of which it is composed. The percentages of carbon, hydrogen, oxygen, nitrogen and sulphur are determined as well as the percentage of ash in dry coal. Such analyses show the carbon contents of coal to vary from about 98 per cent in the graphitic anthracite through about 97 per cent in the semi-anthracite, 87 per cent in semi-bituminous, 80 per cent in bituminous and 74 per cent in lignites to as low as 61 per cent in peats. The corresponding figures for hydrogen run from about 1 per cent through a range in the neighborhood of 5 per cent for semi-bituminous to about 6 per cent in the case of peat.

Oxygen varies from about 2 per cent or less in the case of good anthracite to as high as 33 per cent for peat; nitrogen generally forms about 1 per cent of the dry fuel and sulphur from 1 to 3 per cent.

139. Calorific Value of Coals. The calorific value of coals on a basis of combustible has been shown to vary approximately according to a smooth curve, but the local variations are so great that no generally applicable formula for calorific value has yet been proposed. The formula commonly used is based upon the ultimate analysis and is similar to that suggested as approximately applicable in the case of mixtures of combustibles. It is known as **Dulong's formula,** and is

$$\text{B.t.u. per lb.} = 14{,}600\text{C} + \left\{\begin{matrix} 62{,}000 \\ \text{or} \\ 52{,}000 \end{matrix}\right\}\left(\text{H} - \frac{\text{O}}{8}\right) + 4000\text{S}, \quad (100)$$

in which the letters refer to the weight of the various elements contained in one pound of dry coal.

When an accurate knowledge of the calorific value of a fuel is desired it should be obtained by means of a **fuel calorimeter**. There are many varieties of this instrument, but practically all operate on the same general principle. A known weight of fuel is completely burned within a vessel and the heat liberated is absorbed by water or similar liquid. From measurements of liquid temperatures the heat absorbed by the liquid can be determined, and this with some additions for losses of various kinds must be the heat liberated by the fuel. For details see Chapter XX.

140. Purchase of Coal on Analysis. Until quite recently it was customary to buy coal from the lowest bidder provided the material supplied could be made to give satisfactory results in the plant. Obviously the purchaser knew nothing regarding his purchase, and often bought quantities of ash and moisture at the price of combustible. Now, however, the larger power plants and many of the smaller are buying on the basis of analyses and calorific values as determined in calorimeters.

A certain desirable standard analysis is set and certain variations are allowed from it. Wide variations are penalized by deducting so many cents per ton for each variation of a certain degree, and, finally, outside limits are set for moisture and ash beyond which the fuel need not be accepted. In some cases limits are also set for sulphur.

This is the logical method of purchasing coal in large quantities, and is sure to come into very general use as its advantages become known.

141. Petroleum. This material is obtained from drilled wells and has been found in many widely separated sections of the country. The oil wells of Pennsylvania and neighboring States, of Oklahoma, Texas and California have been the most productive and are hence the most widely known.

Natural petroleum, as it occurs in the United States, is

generally a dark, rather thick, oily liquid with a characteristic odor. It varies widely in composition so far as the compounds contained are concerned, but the variations in ultimate composition, specific gravity and calorific value are comparatively small.

The *ultimate analysis of crude oil* generally shows about 83 to 85 per cent of carbon, 13 to 15 per cent of hydrogen and small quantities of oxygen, nitrogen and sulphur.

The *specific gravity* generally lies between 0.80 and 0.90 and in most cases is nearer the upper figure. It is common practice to express the gravity in terms of the Beaumé scale, an arbitrary scale developed for an instrument known as the Beaumé hydrometer. This device is arranged to float in liquids and measures the gravity by the distance to which it sinks. Various corresponding values of the Beaumé scale and specific gravity are given in Table XII for the region most used in connection with petroleum.

TABLE XII

CORRESPONDING BEAUMÉ READINGS AND SPECIFIC GRAVITIES

Beaumé Reading.	Specific Gravity.	Beaumé Reading.	Specific Gravity.
20	0.9333	34	0.8536
22	0.9210	36	0.8433
24	0.9090	38	0.8333
26	0.8974	40	0.8235
28	0.8860	42	0.8139
30	0.8750	44	0.8045
32	0.8641	46	0.7954

The *higher calorific value* varies between 19,000 and 20,000 B.t.u. per pound and the lower value is generally 1000 to 1500 B.t.u. lower.

Crude oil is sometimes used for fuel, but this is undesirable, for two reasons. First, the crude oil contains many highly volatile constituents which can be distilled

off and which have a high market value in the forms of gasoline and allied distillates. Second, the presence of these highly volatile constituents in the oil makes it more dangerous, as combustible vapors are given off in large quantities at low temperatures and the mixtures formed with the oxygen of the air are often highly explosive.

As a consequence, the material generally sold as fuel oil is a residuum left after distilling off the more volatile constituents of the crude oil. It has practically the same properties as the crude, excepting that dangerous vapors are not given off at so low a temperature.

PROBLEMS

1. A sample of coal gives the following proximate analysis: moisture, 5%; volatile, 4.25%; fixed carbon, 80.75%; and ash, 10%. Determine the percentage of combustible and the percentages of fixed carbon and of volatile in the combustible.

2. What variety of coal is indicated by the values obtained in Prob. 1?

3. The following results were obtained in making a proximate analysis of a sample of coal; moisture, 7%; fixed carbon, 56.7%; volatile, 24.3%; ash, 12%. Determine the percentage of combustible and the percentages of fixed carbon and of volatile in the combustible. What variety of coal is indicated by these values?

4. The ultimate analysis of a sample of dry coal gave the following results: carbon, 79.12%; hydrogen, 4.14%; oxygen, 1.84%; sulphur, 0.92%; nitrogen, 0.74%; ash, 13.24%. Recalculate these values for an ash-free coal.

5. Determine by means of Dulong's formula the upper and lower calorific values of the coal described in Prob. 4.

6. The ultimate analysis of a sample of crude petroleum from which all water was removed gave the following results: carbon, 85%; hydrogen, 13%; sulphur, 1.0%; oxygen, 0.25%; nitrogen, 0.12%; ash (sand and similar material), 0.63%. Determine the upper and lower calorific values by means of Dulong's formula.

CHAPTER XVII

STEAM BOILERS

142. Definitions and Classification. The term boiler is generally applied to the combination of a furnace in which fuel may be burned continuously and a *closed vessel* in which steam is generated from water by the heat liberated within the furnace.

Boilers are classified in many different ways, the more important being given in the following schedule:

Classification of Boilers

(1) According to form	(*a*) Plain cylindrical, (*b*) Flue, (*c*) Tubular, (*d*) Sectional, etc.
(2) According to location of furnace	(*a*) Externally fired, and (*b*) Internally fired.
(3) According to use	(*a*) Stationary, (*b*) Portable (as on trucks, or rollers), (*c*) Locomotive, (*d*) Marine.
(4) According to direction of principal axis	(*a*) Horizontal, (*b*) Inclined, (*c*) Vertical.
(5) According to relative positions of water and hot gases	(*a*) Water tube, (*b*) Fire tube.

Examples of boilers of the different types mentioned are given in subsequent paragraphs.

143. Functions of Parts. It has been shown that there are two essentially different parts in the apparatus commonly known as a steam boiler, the *furnace* and the *boiling vessel.* A simple form of boiler known as a **horizontal, return tubular boiler,** or an H.R.T. boiler, is shown in Figs. 193 and 194 with the two essential parts and their components

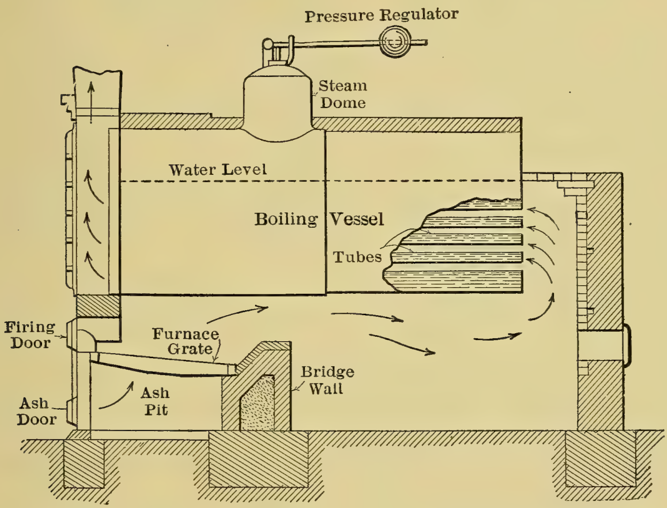


FIG. 193.—Sectional Elevation of H.R.T. Boiler and Furnace.

indicated. The furnace consists essentially of the combination of grates, bridge wall, fire and ash doors, the ash pit and the space above the grates. It is the function of the furnace to so burn the fuel that the maximum amount of heat will be made available for absorption by the water within the boiling vessel.

It is the function of the boiling vessel to transmit to the water within it the greatest possible quantity of the heat thus made available and to resist successfully the

tendency to rupture under the action of the high internal pressure, that is, the pressure of the steam.

In the type of boiler shown the fuel is "fired" by hand, that is, it is spread on the grate by being thrown from a scoop shovel through the opened fire door. Air enters through both doors in regulated proportions and in such quantities as best to approximate complete combustion.

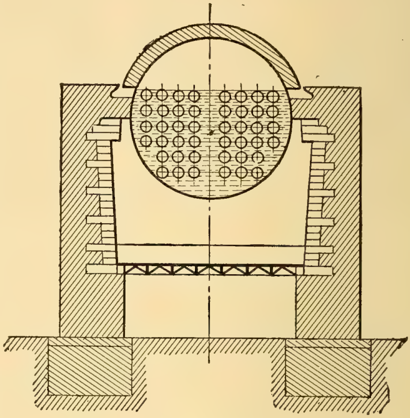


Fig. 194.
Section through Furnace of H.R.T. Boiler.

The hot gases resulting from the combustion pass over the bridge wall, along the lower part of the boiler shell and then through the fire tubes, or flues, toward the front of the boiler as shown by arrows in the figure. From the front end of the tubes the products of combustion pass up through the smoke box to "breechings" or "flues," which carry them to the stack.

Heat is received by the water within the vessel in two different ways:

(1) The hot fuel bed on the grate radiates energy in the same way that the sun or any other glowing body radiates energy. Some of this energy traverses the space between fuel bed and boiler shell and ultimately passes through that shell to the water within. The rest of the radiated energy passes into the walls surrounding the furnace and heats them and the surrounding atmosphere.

(2) The hot gases of combustion pass over the heating surface of the boiler, as shown, and transmit part of their

heat to the water on the other side of those surfaces. The rest of the heat which they carry is either lost to the surrounding walls or is carried up the stack by the gases which leave the boiler at a comparatively high temperature. This temperature ordinarily ranges from about 500° to 700° F. and in extreme cases goes even higher.

144. Furnaces and Combustion. In most forms of boiler the water within the boiler has practically the same temperature as the steam being generated, and this is generally from 320° to 400° F. Obviously the products of combustion cannot be cooled by the water to a temperature below that of the water, so that the gases leaving the boiler in an ideal case would have a comparatively high temperature. Practically, it is found undesirable to attempt to reduce the temperature of the gases to a value even approximating that of the water and, as indicated above, they are discharged at a temperature several hundred degrees higher. In order that the maximum amount of heat may be made available for the boiling vessel the products of combustion must therefore leave the furnace with the highest possible temperature, and the ideal furnace would completely burn the cheapest fuel available in such a way as to give this highest possible temperature and not to generate smoke.

Real furnaces fall far short of this ideal performance, for numerous reasons. The more important of these are given in the following paragraphs:

(*a*) *Incomplete Combustion of Carbon.* In a real furnace the combustion of the carbon of the fuel may be incomplete in two senses; first, some of the carbon may remain entirely unoxidized and pass off with the ash, and second, some of the carbon may be burned to CO instead of to CO_2.

Imperfect combustion of the first kind can result from fuel falling through the openings in the grate before it has been ignited or when only partly burned, or it can result from failure to get air to some of the carbon in sufficient quantities to burn it completely before all of the surround-

ing fuel has been converted into ash and the locality cooled down to such an extent as to allow the unburned carbon in its midst to cool below the temperature of ignition.

Imperfect combustion of the second kind, resulting in the formation of CO, generally results from a lack of sufficient air or from imperfect mixing of air and fuel or of air, fuel and products of combustion. It can also be caused by chilling of air or gases rising from the fuel bed before combustion has been completed. In any coal-burning furnace there is a tendency toward the formation of CO within the bed of fuel, and this tendency is greater the greater the depth of the bed. Such CO burns to CO_2 above the fuel bed when conditions are propitious, the necessary air either passing through the fuel or being admitted at a point above the fuel. It is obvious that if combustion of such gas is to occur in this way, the combustible gas and the air must be brought into intimate contact while they are still at a sufficiently high temperature.

The percentage of CO_2 in the flue gases is commonly taken as an indication of the character of combustion. With pure carbon and theoretical air supply, this should be 21 per cent, and with real coal something between 18 and 20 per cent. Practically, it is so difficult to bring the combustible material and the oxygen of the air into intimate contact that a large excess of air is always used. The excess coefficient in practice varies from about 1.1 to over 2 with averages of about 1.5 under ordinary good conditions. The value 1.5 corresponds roughly to about 13 per cent CO_2 in the flue gases and an excess coefficient of 2 corresponds roughly to 9 to 10 per cent CO_2. Even with such excess coefficients as those indicated as averages, it is not at all uncommon to find a small quantity of CO in the flue gases. The fireman's task with respect to the combustion of carbon thus reduces itself to the use of such a quantity of air in such a way that the minimum loss results from excess air and unburned CO.

(*b*) *Incomplete Combustion of Hydrocarbons.* The hydrocarbons which appear as volatile matter in the proximate analysis are practically all distilled from the fuel, as it is heated in the furnace before ignition in the same way as when making a proximate analysis. If they are to be completely burned they must be mixed with the requisite quantity of air after distillation and both the vapors and the air must be maintained at a sufficiently high temperature until combustion is complete. The air for the combustion of distilled volatile may all filter through the fuel bed or some of it may be admitted at a point above the level of the fuel bed.

If the flame formed by burning hydrocarbons is allowed to come in contact with cold surfaces, as, for instance, the heating surfaces of the boiler, the gases are cooled below the temperature of ignition and combustion ceases. This results in the deposit of soot (unburned carbon) upon the heating surfaces of the boiler and in the carrying of soot and unburned hydrocarbons up the stack. The soot and some of these hydrocarbons form the unsightly smoke so familiarly associated with some stacks.

Or, if the air supply is at a sufficiently high temperature, but is insufficient in quantity, the hydrocarbons are incompletely burned and smoke results.

The formation of smoke can be conveniently studied by means of the ordinary kerosene lamp. Such a lamp operates by burning hydrocarbons of the same general character as those distilled from solid fuels. The hydrocarbons are drawn up by the wick in the form of liquids, are vaporized by heat near the top of the wick and then combine with oxygen from the atmosphere to give the luminous kerosene flame.

If the flow of kerosene and the air supply are properly adjusted and if the temperature is high enough, the combustion results in the formation of invisible and practically odorless gases. If, however, the air supply be decreased

or be greatly cooled, a very smoky and very odorous combustion ensues. The same result could be obtained by the use of too great a quantity of air, a condition often attained when the supply of kerosene in the bowl of the lamp is almost exhausted.

The effect of a cold surface is easily seen by inserting a cold metallic or porcelain surface into the tip of the flame and then withdrawing it. It will be found covered with soot.

(*c*) *Advantages and Disadvantages of Excess Air.* It has been shown that excess air is practically necessary in the real furnace in order to insure against a deficiency at any point, and it is thus advantageous in that it makes the combustion more nearly complete than would otherwise be the case. On the other hand, excess air represents just so much excess material to be heated at the expense of heat liberated by combustion and hence decreases the maximum temperature attained. A sufficiently great supply of excess air could so reduce the temperature that even if combustion were complete very little heat would be made available for absorption by the boiling vessel, because the temperature attained by the products of combustion would be too low.

Excess air in large quantities may also result in cooling unburned gases before combustion to such an extent as to make the completion of combustion impossible.

145. Hand Firing. The commonest type of furnace is that shown in Figs. 193 and 194, and the commonest method of hand firing consists in spreading a layer of fuel as evenly as possible over the entire surface of the fuel bed as often as required to replace the fuel burned away. At such intervals as experience shows to be necessary the fire is cleaned, that is, the ashes are worked out from under the fuel by means of slice bars, so that practically nothing but live fuel resting on a thin layer of ash remains behind.

This method is open to many serious objections; the more important are;

1. There is a gradual increase in thickness of fuel bed from the time of one cleaning until the time of the next. This gives a constantly changing set of requirements for the proper proportions of air entering below and above the fuel bed and a constantly changing resistance to flow of air through the bed, so that great skill is necessary if the best conditions are to be maintained throughout.

2. There is always a tendency for a fuel bed to burn faster at some points than at others, due to the accidental distribution of fuel, ash and air. Where "holes" are formed in this way large quantities of comparatively cold air can pass through with the consequences already enumerated. It takes considerable skill and watchfulness on the part of the fireman to prevent the formation and continued existence of such holes.

3. The firing door must be opened wide every time that fuel is to be fired, that is, at intervals varying from two or three minutes to fifteen or more, depending on load, character of fuel, etc. While the door is open large quantities of cold air readily flow into the furnace and cool down all parts of it, and a proportionately smaller amount will ordinarily pass through the fuel bed. The result of this on the flue gases and operation of the boilers has already been considered, but there is another result of equal or greater importance. As a consequence of this action the volatile hydrocarbons distilled off from the freshly fired fuel, which are themselves at a comparatively low temperature, are surrounded on all sides by cooled walls and come in contact with cold air only. The chances of their burning completely are very slight, and a great part of these volatilized materials passes off unburned as invisible gas and as smoke. Obviously the greater the volatile content the greater the difficulty, so that anthracite causes least trouble in this way, while most bituminous coals give heavy black smoke when burned under these conditions.

The cooling down of the interior of the furnace during

firing is accompanied by the covering of the fuel bed with cold fuel, so that, for the time being, very little radiant heat enters the boiling vessel, and the gases which come in contact with its surface are comparatively cool. The maintenance of a constant steam pressure under these conditions is practically impossible, but the difficulties can be partly overcome by very frequent firing of small quantities, so that the door is open a very short time and also that the layer of fuel is very thin and does not cut off much heat.

4. The cleaning of the fire necessitates keeping the fire door open for several minutes, with results of the same variety as those just enumerated.

Summing up these difficulties, they divide themselves into two classes—those which can be almost or entirely eliminated by skill of a very high order and those which are inherent and cannot be eliminated by skill. It will also be observed that all should give more trouble with fuels high in volatile than with those of the anthracite variety, both as to incomplete combustion and to the formation of smoke.

Several other methods of hand firing have been proposed, particularly for use with bituminous coals, and some of them have been successfully utilized in isolated instances. Nearly all depend upon covering only part of the fuel bed at one time and, by alternating the parts covered in this way, fresh fuel on one part of the bed is coked while air is heated by coming in contact with the uncovered incandescent part of the bed and is therefore in proper condition to burn more perfectly the volume of hydrocarbons being distilled off. These methods are all good, but they involve a great deal of careful work and a high degree of skill on the part of the fireman.

Other methods of eliminating some of the difficulties depend upon modifications of the furnace and air supply. Most attempt to entirely surround the fuel and the gases given off with heavy masses of brick work and tile, so that

enough heat will be stored during incandescent periods to tide over the periods of cooling. Some forms have combined with this idea a series of air ducts in the brick work so arranged that air on its way to the furnace passes through these ducts and is heated. In some cases the air supply is automatically controlled and more air is supplied above the fire during the period of distillation, or coking, as it is called, than during the following period, when the coked coal is brightly incandescent and little volatile matter is present.

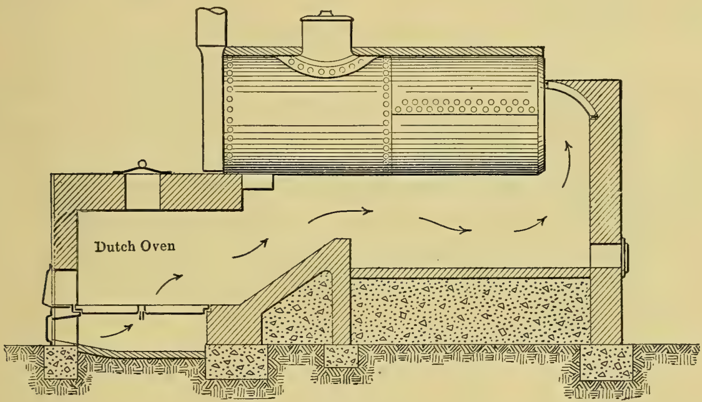


Fig. 195.

In some hand-fired furnaces which are intended for use with bituminous coals that give a long flame the parts of the boiling vessel within range of the flames are covered with tiles. This prevents impingement of unburned gases upon cool surfaces and thus tends to prevent the formation of smoke and incomplete combustion.

Carrying this principle to its logical conclusion results in the installation of the grate in a firebrick chamber in front of the boiler-setting proper, as shown in Fig. 195. Such a device is known as a *Dutch oven* and is often very efficient in totally or partially preventing the formation of

smoke. It does not, however, give as high an economy as might be expected, because a great part of the radiant heat of the fire does not reach the boiler surfaces and because the large external surface results in great radiation losses to atmosphere.

Another interesting modification consists of reversing the direction of the draft, that is, the direction in which the air passes through the fuel bed. The type of furnace already described is known as an **updraft furnace,** because the air passes upward in flowing through the bed. The modified type here referred to is called a **downdraft furnace,** because the air flows downward in passing through the fuel.

In downdraft furnaces the coal is fired on top of the grate as in other types, but the air is admitted above, flows downward toward what would normally be the ashpit, and from there on over the heating surfaces of the boiler. Fresh coal fired on top of the incandescent bed in such a furnace distills as in other types, but the volatiles are mixed with the entering air and are carried downward through the hot bed so that ideal conditions for combustion are more nearly attained. In some forms there is a second updraft grate beneath the downdraft grate. This second grate receives partly burned coals falling through from the upper grate and holds them until combustion is practically completed.

In downdraft furnaces the grate bars are generally made of pipes, and water, from the boiler or on its way to the boiler, is circulated through them. If this were not done the grates would quickly warp out of shape and ultimately burn away because of the high temperatures to which they are subjected.

146. Mechanical Grates. In order to overcome the difficulties arising from opening the doors for the purposes of cleaning the fire, numerous so-called rocking, shaking, self-cleaning, or dumping grates have been developed. These are generally built up of grate bars which have a

rough T or an inverted L section with the upper horizontal branch of the T or inverted L slightly rounded, as shown in Fig. 196. These bars are arranged in groups with their longitudinal axes running across the grate, and they are so supported that they can be rocked about a point in the vertical leg of the T or L by means of levers located at the front of the boiler. By rocking the bars the lower part of the fuel bed which has been burned to ash can be dropped into the ash pit, while the upper part is sufficiently agitated to close up holes which may have formed, and this can all be done

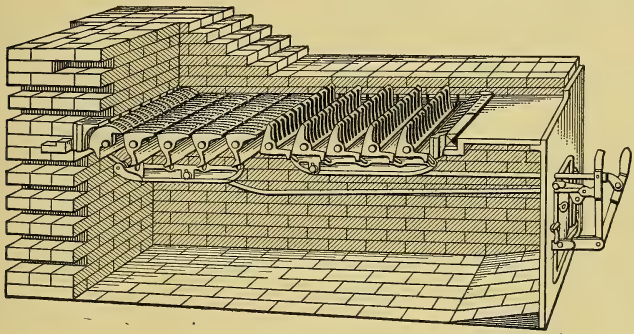


FIG. 196.

with the doors closed. Or, if desired, part or all of the fuel bed can be dropped into the ash pit by a similar rocking motion.

147. Smoke and Its Prevention. An idea of the reasons for the formation of smoke will have been obtained from the preceding paragraphs. A reasonably skillful fireman should have little difficulty in burning anthracite coals in the simpler forms of furnaces without smoke, but it is almost impossible to commercially burn many of the varieties of bituminous coals in this way without the formation of excessive volumes of dense black smoke at intervals immediately following each firing.

Aside from all æsthetic and sanitary considerations, smoke is undesirable because it represents poor furnace conditions and waste. The actual loss of carbon in visible smoke is generally almost negligible in comparison with the other losses in the form of unburned hydrocarbons, the lowered initial temperature, etc. All of these losses combined represent a waste of considerable magnitude.

The proper method of smoke elimination is not the combustion or removal of smoke already formed, but it is the burning of fuels in such ways as not to form any appreciable quantity in the first place. To accomplish this end the following must be achieved:

1. Coal must be fired continuously and uniformly without the opening of doors which admit cold air to the furnace.

2. Volatiles must be distilled continuously and uniformly and in such a place that they are given ample opportunity to mix with proper proportions of air and to burn completely before coming in contact with cool surfaces.

3. The air supply must be properly controlled and tempered to meet the demands of the fuel both in and above the bed.

4. The fire bed must be worked continuously and uniformly so as to eliminate ashes as rapidly as formed and to maintain a bed of uniform depth and condition.

Some of these necessary conditions can be attained by the use of the various forms of hand-fired furnaces already described but, even in the hands of skillful and industrious men, it is impossible to meet all of them. *Mechanical stokers* which more nearly approach the ideals set have therefore been developed and are widely used.

148. Mechanical Stokers. These mechanical devices are useful for two reasons—they eliminate a great deal of labor and they make possible the burning of many varieties of refractory fuels without the formation of excessive quantities of smoke.

Despite the good results which can be achieved by their use, mechanical stokers are not installed in small plants as often as might be expected. This is because good stokers are very expensive in comparison with hand-fired furnaces and, despite economy of fuel, do not generally show a financial saving unless their use eliminates the services of several firemen.

It is generally assumed that one man can care for water, coal and ashes for about 200 boiler horse-power or can handle coal only for about 500 boiler horse-power. Experience has shown that one man can fire about 2000 to 5000 boiler horse-power when the boilers are equipped with good stokers and coal-handling apparatus.

Financial calculations will not justify mechanical stokers in many of the smaller plants in which they are used. However when improved conditions with respect to smoke and dirt and improved labor conditions are taken into account they are generally regarded as paying propositions.

Mechanical stokers can be roughly divided into two types, those which duplicate hand spreading of fuel and are known as **sprinkler stokers,** and those which supply fuel at one or more points and work it progressively toward the ash end of the apparatus as it burns. The first type has not been widely installed, though it is possible that it may meet with more popular approval after further development.

Stokers of the second type may be roughly divided into four classes, which are

1. Chain grates.
2. Inclined stokers or overfeed stokers.
3. Underfeed stokers.
4. Combinations of above.

A **chain grate,** as made by the Illinois Stoker Company, is illustrated in Figs. 197, 198, 199, and 200. It consists

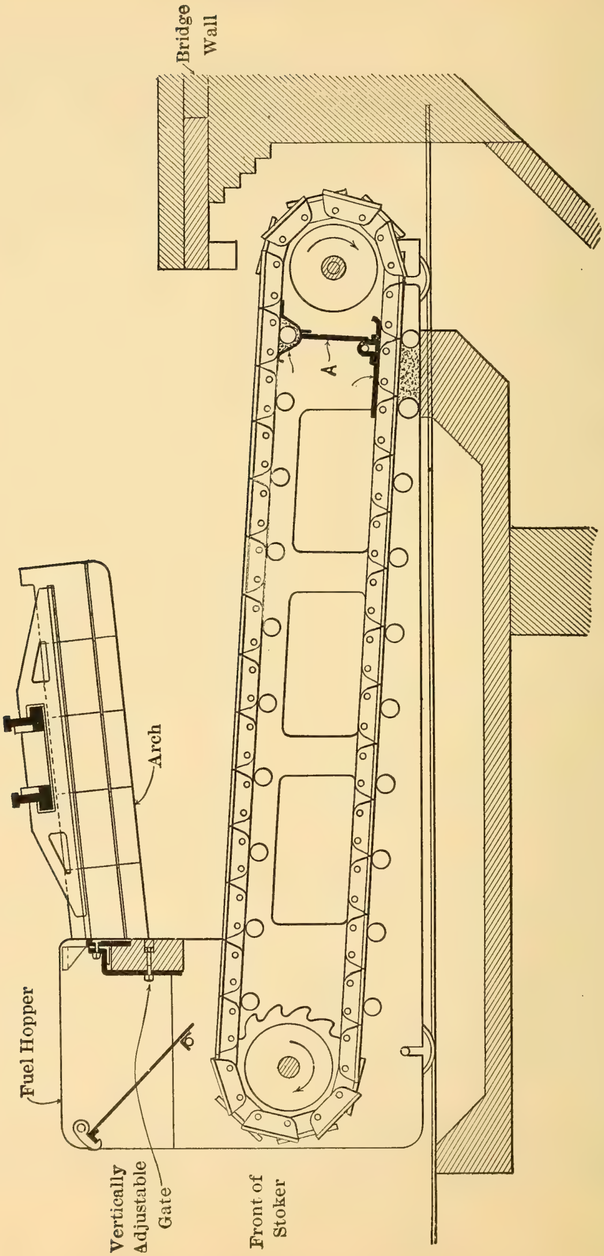


FIG. 197.—Illinois Chain Grate Stoker.

of a broad chain made up of a great number of small links and carried on toothed wheels and roller wheels supported in a frame which can be wheeled into position within the

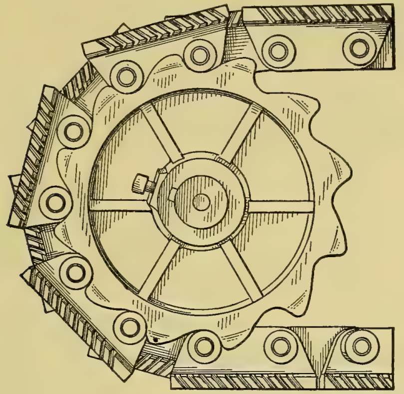
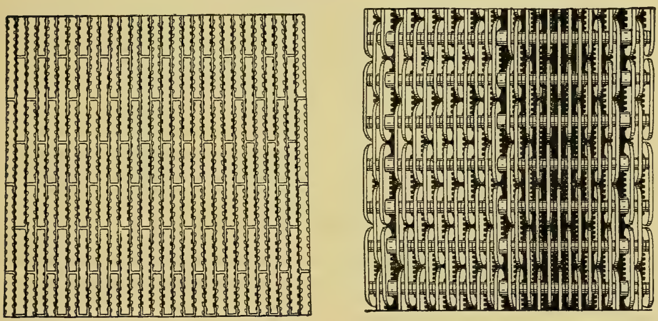


FIG. 198.—Sprocket and Links of Illinois Chain Grate.



TOP VIEW OF CHAIN
SHOWING DISTRIBUTION OF AIR SPACES

BOTTOM VIEW OF CHAIN
SHOWING ROLLERS FOR DRIVING-SPROCKET ENGAGEMENT

FIG. 199.

boiler setting. The general arrangement of the chain and rollers is shown in Fig. 197; details of the front or driving rollers and of the links are shown in Fig. 198; a top and bottom view of part of the chain is given in Fig. 199; and Fig. 200 is a perspective view of the frame showing the

tracks on which it may be rolled into and out of the boiler setting.

The chain is driven slowly in the direction indicated by the arrows in Fig. 197 by power applied, through worm gearing, to the shaft of the toothed wheels at the front of the stoker. Coal feeds automatically from the hopper by gravity and is carried into the combustion space by the moving chain, the thickness of the bed being controlled

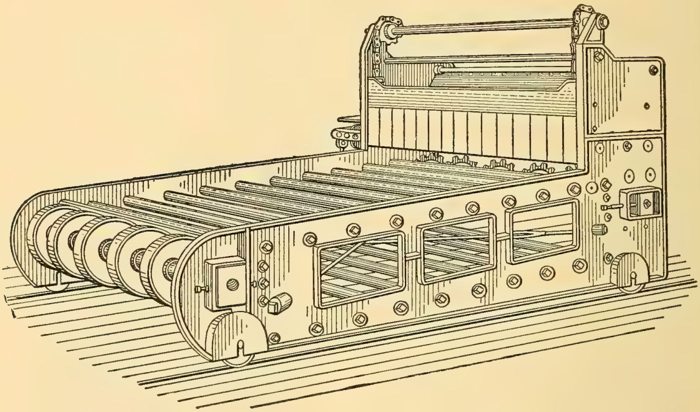


Fig. 200.—Framework of Illinois Chain Grate.

by the height of the adjustable gate shown. As the fuel enters the furnace it passes under the coking arch, which spans the entire front part of the grate and which is maintained at a high temperature by heat radiated from the incandescent fuel nearer the inner end of the grate. The volatiles are distilled from the fresh coal by heat received from this arch and are heated and mixed with air at this point. The coked fuel is then carried on into the furnace and burned, the refuse being discharged at the bridge wall.

If the thickness of bed and speed of chain travel are properly adjusted, all of the fuel can be coked before passing out from under the arch and can be burned almost

completely before reaching the bridge wall, so that practically ashes only will be discharged.

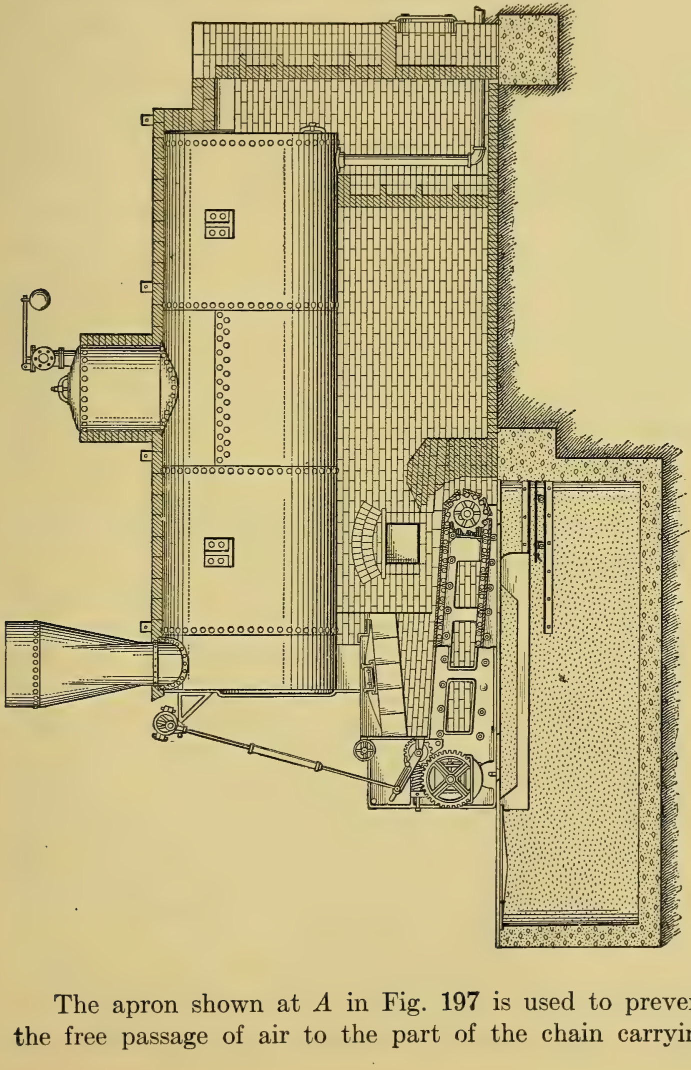


Fig. 201.—H.R.T. Boiler Fitted with Chain Grate.

The apron shown at *A* in Fig. 197 is used to prevent the free passage of air to the part of the chain carrying

practically nothing but ash, as this would result in excessive dilution of the products of combustion.

A stoker of this type installed under a horizontal return-tubular boiler is shown in Fig. 201. In the illustration part of the side frame of the stoker is broken away in order to show the chain and its roller guides. The eccentric shown near the top of the front of the boiler drives the chain through an arm of adjustable length, which makes possible the control of the speed of chain travel.

The early forms of chain grates were intended for use with natural draft, that is the space under the grate connected directly with the atmosphere and maintenance of a " draft " or " under-pressure " in the furnace permitted the external atmosphere to push through the fuel bed, this supplying the necessary oxygen for combustion. As the art developed a demand arose for stokers capable of burning more fuel per square foot of grate area. This demand was met by developing stokers which could be used with " forced draft." These stokers were so arranged that air could be forced through the fuel bed by fans discharging into the space beneath the stoker.

The better designs of forced draft, chain grates provide for differential air supply to different sections of the fuel bed. For this purpose the space between the upper and lower chains is divided into compartments by vertical partitions in planes at right angles to the travel of the grate. Provision is then made for controlling the air pressure in each box separately by means of dampers or equivalent. With such construction the air pressure applied to the under surface of the grate can be regulated by sections instead of for the entire grate and it becomes possible to grade the air supply to more nearly meet the requirements of the fuel in different stages of combustion.

As an example of the usefulness of such an arrangement, consider the last section over which the material passes before being discharged from the end of the stoker. If

the fuel is very nearly burned out before it reaches this section a very small amount of air will suffice for the completion of combustion. Any greater amount would simply pass through without useful effect, would serve to lower the furnace temperature and to increase the excess coefficient. On the other hand if the fuel still contains large amounts of unburned carbon when it reaches this section it is desirable to be able to supply a relatively large amount of air so as to more nearly approximate complete combustion of the carbon before the material is discharged from the grate.

Development of the forced draft chain grate and of numerous small but important refinements in details of design has brought the chain grate into particular prominence in connection with the combustion of certain very poor grades of fuel. Thus very fine sizes of anthracite previously considered of no commercial value because of their small size and high ash and moisture content are successfully burned on chain grates particularly designed for such use. Some of the poorer varieties of bituminous fuel with high ash content of such character as to form excessive amounts of clinker, are also being burned on such devices.

An inclined stoker with front feed and a step grate, known as the Roney stoker, is shown in Figs. 202 and 203. The fuel is fed out of the hopper and onto the dead plate by means of the reciprocating pusher. From the dead plate it is pushed down upon the grate bars by the following fuel. These bars are rocked mechanically so that their tops alternately assume horizontal and inclined positions, and this action feeds the fuel downward until it is discharged onto the dumping grate. The material collecting on this grate is periodically dropped by hand into the ashpit.

The fuel is coked while passing under the coking arch and the coked material is practically completely burned by the time it has traveled down the grate. The volatiles

are mixed under the coking arch with heated air which has passed through the grate and with heated air forced in above the fuel.

An inclined stoker of the side-fees type with bar grates known as the Murphy stoker, is illustrated in Figs. 204, 205 and 206. This stoker is provided with two coal-

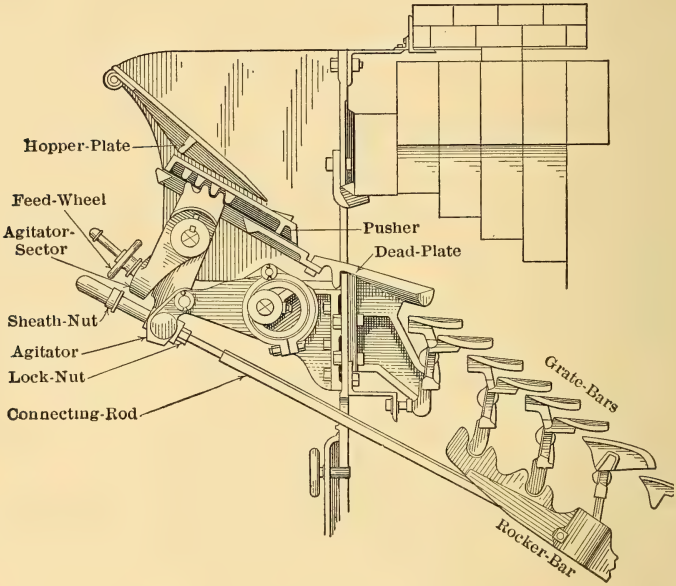


Fig. 202.—Details of Feed Mechanism, Roney Stoker.

magazines or hoppers which are placed horizontally in the side walls of the boiler setting and feed fuel onto the inclined grate bars, Fig. 204, which carry it downward toward the lower point of the V formed by the grates. The grate bars, Fig. 206, are alternately fixed and movable, the movable bars being hung from above and their lower ends being moved up and down by power furnished by a small steam engine or other convenient source.

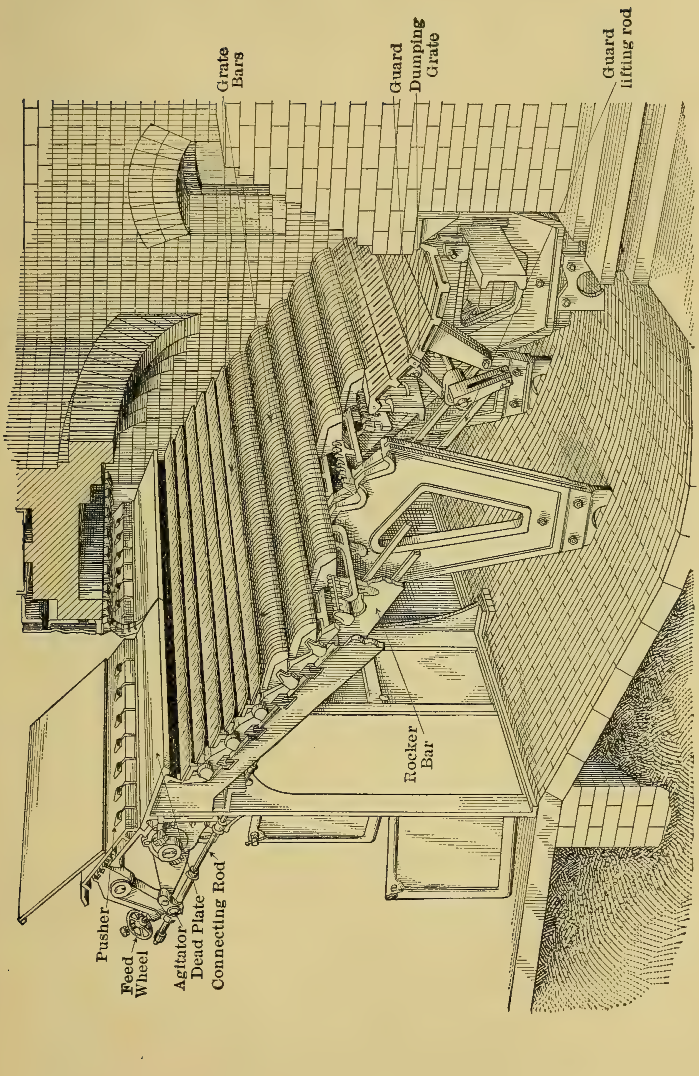


FIG. 203.—Roney Mechanical Stoker.

A toothed bar arranged for rotation by hand or by power is located at the bottom of the *V* and is used for grinding up ash and clinker which is too large to fall through into the ash pit. This bar is kept cool by making it hollow and connecting one end to the smoke flues or stack so that air is constantly drawn through it.

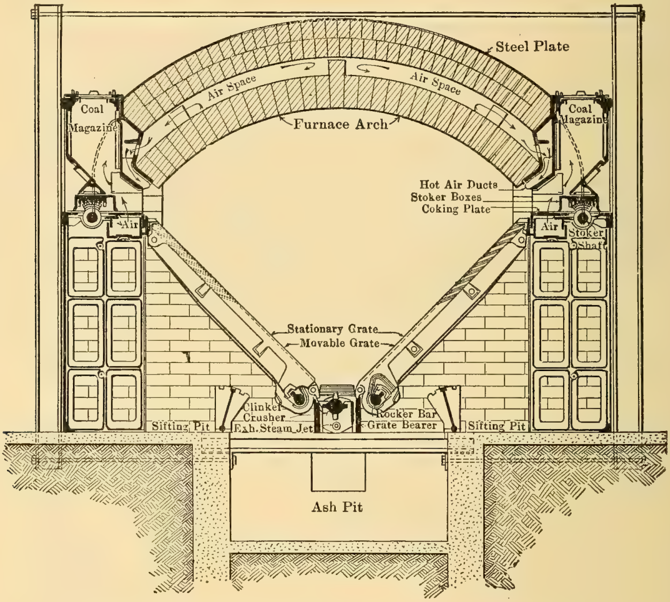


FIG. 204.—Transverse Section of the Murphy Stoker.

The location of the coking arch and the method used for supplying warm air should be evident from the figures.

A stoker of this variety is shown in place under a horizontal water-tube boiler in Fig. 207.

An underfeed stoker made by the Combustion Engineering Company is shown in Figs. 208 and 209. Coal is fed from the hopper onto the reciprocating bottom *B* by means of the reciprocating pusher *P*. The part *B* forms the bottom of a trough as shown in Fig. 209, and its reciprocating motion feeds the coal upward and out of this trough so that

it spills over onto the inclined grate bars. The reciprocating motions are all obtained from the direct-acting steam cylinder shown.

The inclined grate bars are alternately fixed and movable, the movable bars sliding back and forth at right angles

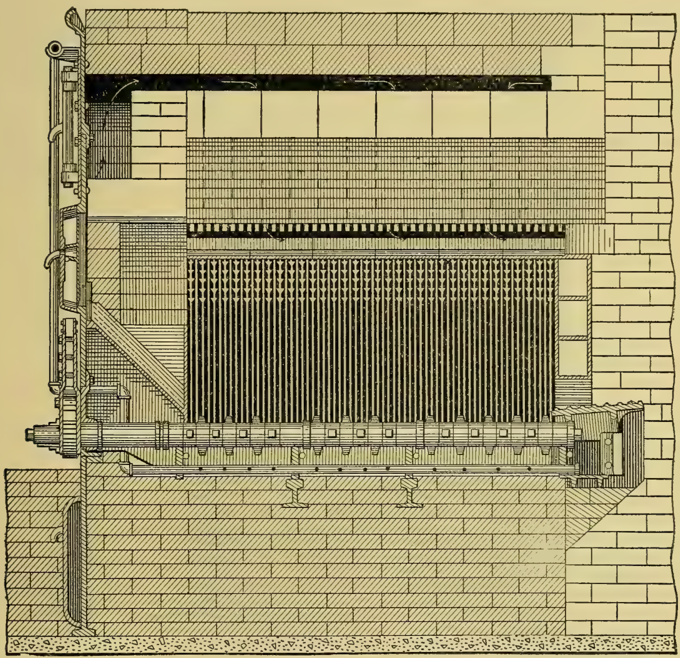


FIG. 205.—Longitudinal Section, Murphy Stoker.

to the trough under the action of horizontal rocking bars *R*. This action gradually feeds the fuel downward and toward the side of the furnace, the refuse finally landing on the dumping trays shown.

Air enters the duct below the trough through the adjustable gate *G*, controlled by crank *C*, and part of it passes out through holes *H* hear the top of the trough,

Fig. 209. The remainder passes down through the hollow grate bars and into the heated air box from which it flows upward between the grate bars.

It will be observed that the coal is fed onto the grate from below, so that all volatiles distilled off must pass upward through the incandescent fuel before entering the space above the fuel bed. Part of the air which is to burn

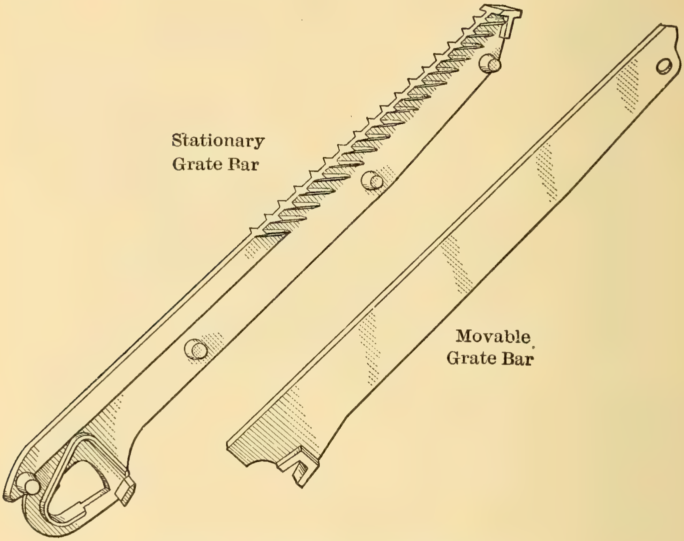


Fig. 206.—Grate Bars of Murphy Stoker.

this volatile matter also passes through the fuel bed and the remainder flows over the incandescent fuel from the opening shown near the hopper in Fig. 208. The air and volatiles are thus raised to a high temperature and well mixed, and the operation is continuous and uniform, all tending to facilitate smokeless combustion.

Another variety of underfeed stoker known as the Taylor stoker is shown in Fig. 210 (*a*), (*b*) and (*c*). This stoker is built up of alternated retorts and air

boxes, the proper number to give the desired width of stoker being used. Coal is fed from the hoppers

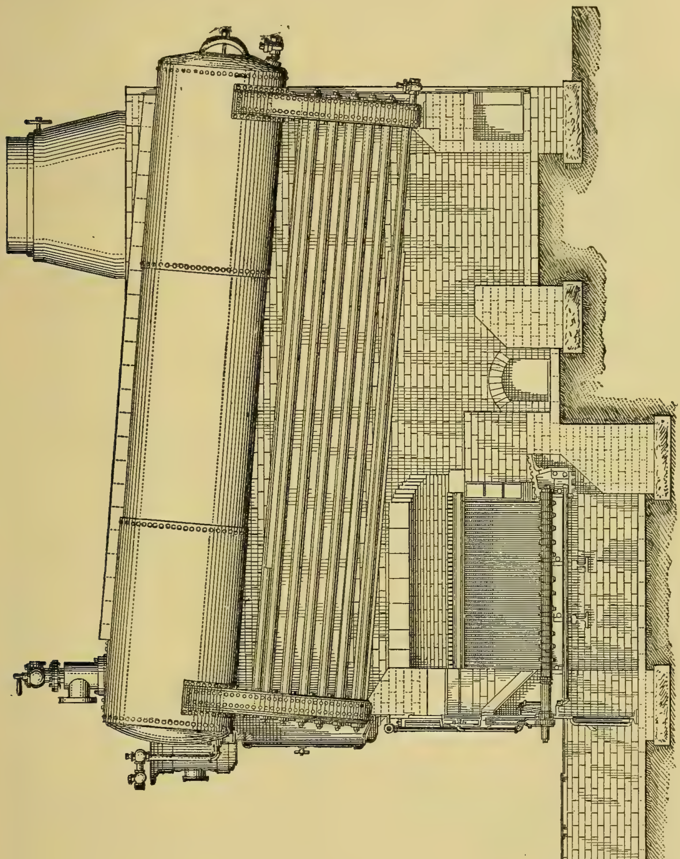


Fig. 207.—Murphy Furnace Arrangement as Applied to Horizontal Water-tube Boiler.

into the retorts by the upper ram or plunger shown in Fig. 210 (*b*) and part of it is again pushed forward by

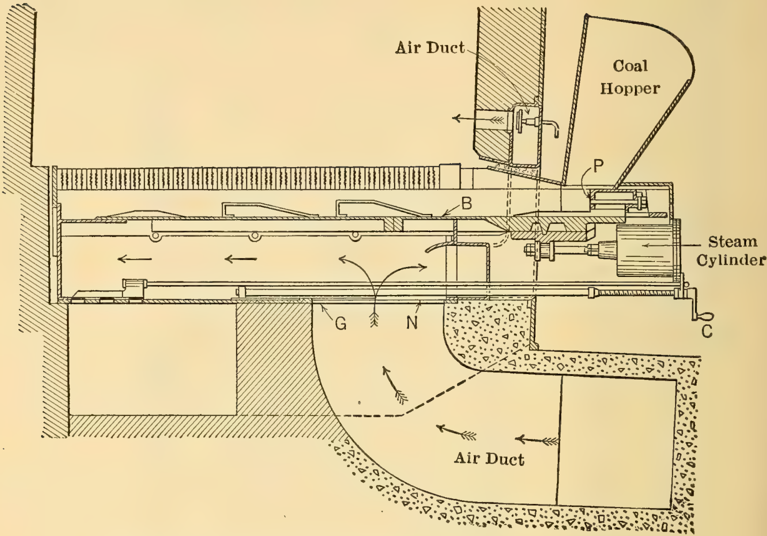


FIG. 208.—Longitudinal Section of Type "E" Stoker

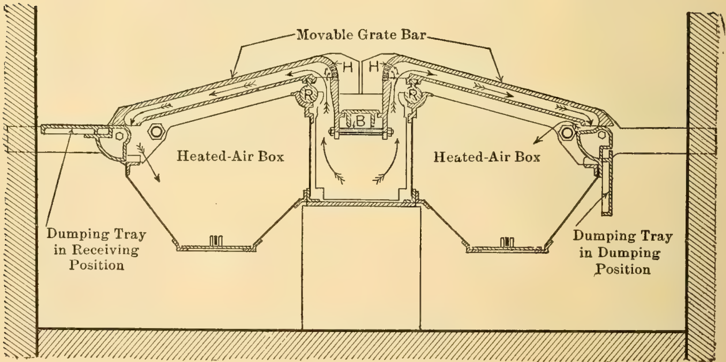


FIG. 209.—Cross Section of Type "E" Stoker.

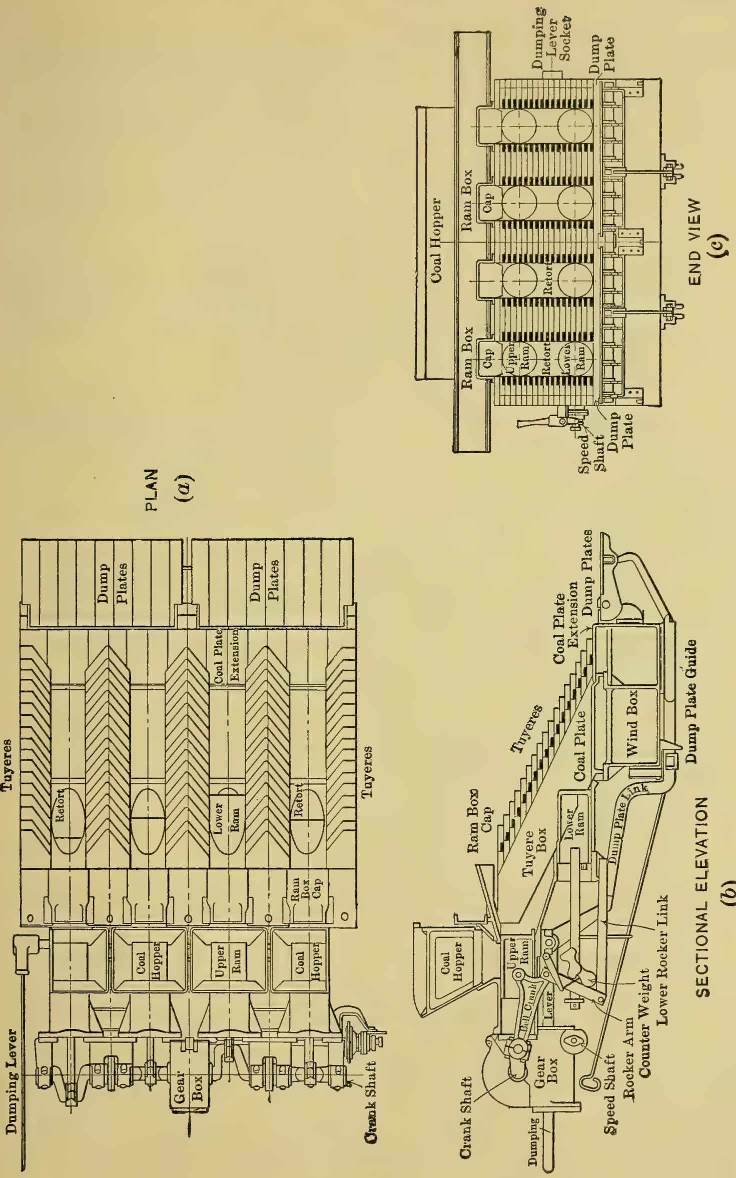


FIG. 210.—Taylor Stoker.

the lower ram or plunger. The stroke of the lower plunger can be regulated and in this way the relative quantities of coal pushed forward in the upper and lower parts of the retorts can be controlled. The coal spreads over the tuyere blocks which form the inclined tops of the air boxes and forms a comparatively even, inclined layer of fuel.

Coking proceeds under the incandescent fuel which forms the upper surface of this layer, and the volatiles mix with air entering through the hot tuyeres and pass upward through the hot fuel above.

In this stoker advantage is often taken of the fact that the draft (pressure of air) required with underfeed stokers is so great that it can be more economically attained by the use of a fan than by the use of a stack. The fan and the coal-feeding plungers are both connected to one engine and the speed of this engine is automatically controlled by the steam pressure within the boiler. As this pressure decreases the engine speeds up, thus delivering more coal and air and as the pressure increases the engine slows down with opposite results. By properly fixing the travel of the plungers initially, the best relative proportions of air and coal are set for the entire range of loads to be carried and the variation of both is thereafter in approximately the same proportions.

A stoker of this type in position under a horizontal water-tube boiler is shown in Fig. 211. A double-ended arrangement of Taylor stokers as used under very large water-tube boilers is shown in Fig. 212.

Powdered or **pulverized coal** burning equipments have been invented in great numbers and are successfully used in several of the industries. They are essentially stokers. Many attempts to use pulverized coal for firing boilers have been made from time to time but until quite recently the results were not considered satisfactory. Within the past few years a number of installations have been made in

boiler plants and the results are claimed to be satisfactory in many instances.

The coal after crushing to moderate size is dried by passing it through a drier heated by hot products of com-

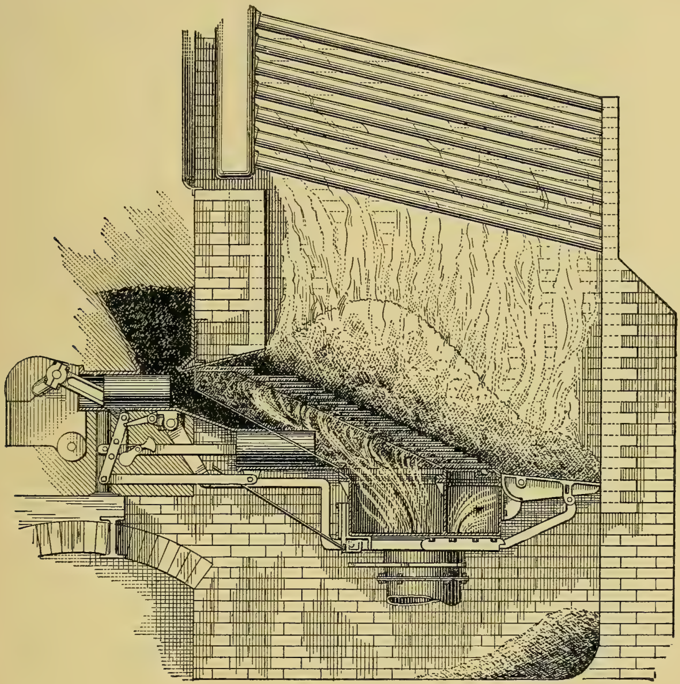


Fig. 211.—Taylor Stoker Under Horizontal Water-tube Boiler.

bustion unless the moisture content "as received" is not considered too great for satisfactory pulverization and utilization. The material is then pulverized in a mill arranged to give very fine subdivision, pulverizing so that at least 60 to 90 per cent passes through a 200 mesh sieve being common practice.

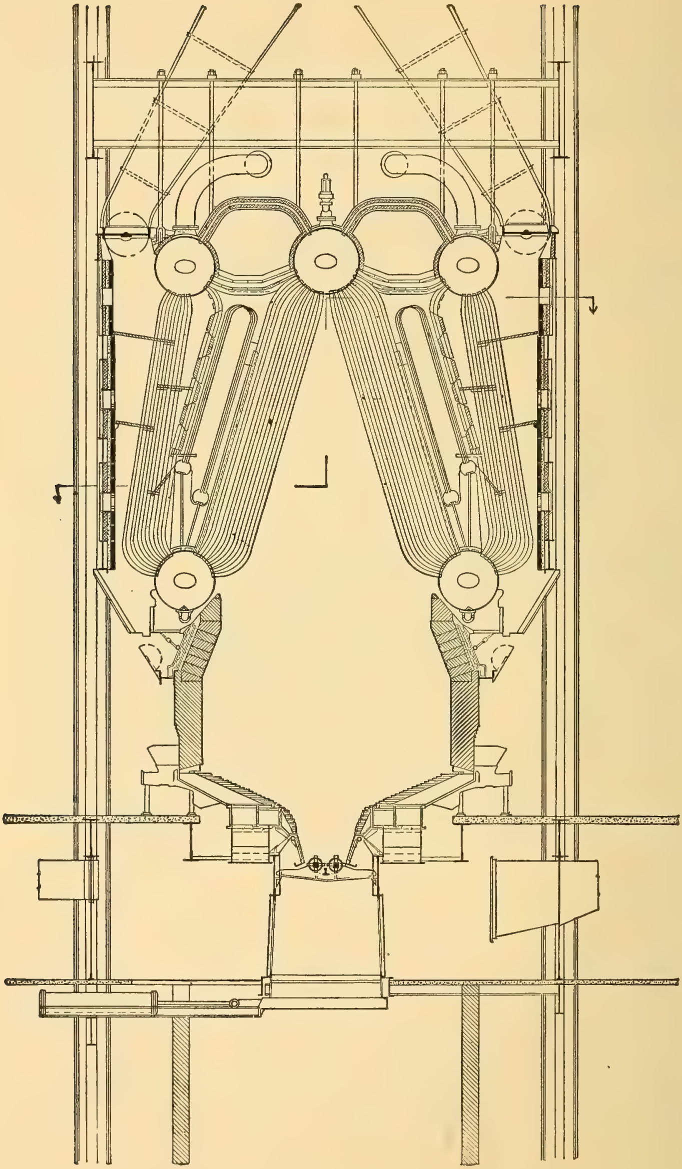


Fig. 212.—Double-ended Arrangement of Taylor Stoker under Sterling Type W. Boiler.

The pulverized fuel is then transported to small hoppers immediately adjacent to the individual boilers. From these hoppers it is fed into a stream of air in which it becomes suspended and by which it is carried through the burner into the furnace. The burner generally consists of a simple nozzle. The fuel burns in the form of a torch at the end of the nozzle, combining with the air which carried it in and generally with additional air admitted through controllable doors or openings in the walls of the furnace.

The fine pulverization makes possible very rapid combustion of the individual particles and the flames have the appearance of an intensely hot gas flame or liquid fuel flame. Success depends on completing combustion while the particles of fuel are still suspended in space, that is before they have had time to settle or to impinge on any solid surface. The flame is easily brought to so high a temperature that the ash contained in the fuel is liquefied. Unless the installation is carefully planned and properly operated this fused ash is very apt to cause serious difficulties by building up stalagmites and stalactites in the furnace or by actually solidifying on the cold heating surfaces of the boiler.

Many advantages are claimed for pulverized fuel firing in boiler plants. The principal claims are great flexibility with respect to character and quality of fuel burned, ease of operation so that one fireman can handle a large boiler installation, ease and simplicity of regulation so that high thermal efficiency can be obtained and maintained even under rapidly changing conditions, and elimination of banking losses as fuel is entirely excluded from the furnace when steam is not desired. However, the use of pulverized fuel entails a large investment in equipment for preparing it and the fixed charges on this equipment as well as the operating expenses chargeable to preparation tend to balance gains resulting from the advantages

enumerated above. The art is as yet so young that it is impossible to obtain sufficient data to make a true and complete comparison between pulverized and solid fuel firing.

Oil firing is essentially a mechanical, rather than a manual process, and while oil burners are not ordinarily understood as belonging to the class of mechanical stokers, they have all the essential characteristics of such apparatus.

To burn oil successfully under a boiler it must be finely atomized and mixed with the necessary quantity of air, and there must be sufficient open space within the furnace for the free development of the flame and the completion of combustion before impingement on cool surfaces.

Oil-burning furnaces are generally given a rather large volume; considerable firebrick is used in such ways as to give incandescent walls and baffles to assist ignition and combustion, and all heating surfaces are arranged so that they are not in the direct path of the flame.

The atomization of the oil is effected in two distinctly different ways. In some forms of burners it is brought about by mechanical means, the oil being pumped through a nozzle of some sort which is so shaped that the issuing jet breaks up into a great number of very small particles. In other forms, steam is used to break up the jet, the steam and oil entering the body of the burner separately and later coming into contact in such a way that the oil is literally torn apart by the steam. This form of burner has been more extensively used in the United States than has the former but present developments indicate a probable reversal in this respect. The mechanical type of burner has been developed to a very high degree during the past few years, and one form, at least, seems to possess marked advantages.

Oil burning shares with the burning of powdered coal,

the property of permitting very accurate regulation of the air supply to suit the quantity of fuel being burned. The excess coefficient may therefore be maintained at a low value and the initial temperature may be made correspondingly high. Part of the advantage thus gained over the commoner methods of coal firing is, however, counterbalanced by the quantity of steam used for heating and pumping the oil and for atomizing in some forms of burners.

Both oil burning and powdered-coal burning can be easily made to give smokeless combustion in properly designed furnaces and both yield readily to forcing. That is, the temporary consumption of excessive quantities of fuel to tide over short demands for excessive amounts of steam is comparatively easily effected if sufficient furnace volume is available.

149. Rate of Combustion. The rate at which coal is burned in a given furnace or on a certain grate is generally given in terms of *pounds of coal fired per square foot of grate surface per hour* and is referred to as the rate of combustion.

The rate at which coal can be consumed is largely dependent on the intensity of draft available, that is, on the air pressure available for driving air through and over the bed of fuel. The higher the pressure available, the greater will be the quantity of air which can be supplied and the greater will be the quantity of coal that can be burned. If it were not for the cost of creating the draft, the only limit to increasing the rate of combustion would occur when the velocity of the air became so great that the fuel would be picked up from the grate and carried onward into the flues in a partly burned condition. Commercial drafts give pressure differences above and below the fuel bed which range from about 0.1 inch of water to as high as 8 ins. In stationary plants the pressures generally range from 0.1 to about 0.5 in cases

where hand firing is employed, and are carried as high as 5 or more inches of water with some forms of mechanical stokers.

The best rate of combustion varies with the type and size of fuel, the type and size of furnace, the type and size of boiler, the draft and many other considerations. In ordinary power-plant practice with hand firing the rates of combustion commercially used generally fall within the following limits: with anthracite, 15 to 20 lbs. per square foot per hour; with semi-bituminous, 18 to 22 lbs.; and with bituminous, 24 to 32 lbs. In the case of stokers, these values may be doubled and even trebled if proper provisions are made.

As practically all of the volatile is consumed above the grate, the fixed carbon content is practically the determining factor, since it is this constituent that is burned on the grate. This explains the high rate possible with fuels with high volatile content. The most economical results are generally obtained when from 12 to 16 lbs. of fixed carbon are consumed per square foot of grate per hour.

The figures given above do not represent limiting conditions. In torpedo-boat practice, where high-draft pressures are used (from 4 to 8 ins. of water), rates of from 50 to 120 lbs. are attained. On locomotives, which also use high-draft pressures, rates of combustion greatly in excess of stationary practice are generally used.

The capacity of a given boiler, that is, its ability to generate steam, increases as the rate of combustion is increased, since more heat is thus made available. The economy of the combination, that is, pounds of steam generated per pound of coal fired, increases until some best rate of combustion for the fuel in question is reached, and thereafter decreases. The variation of economy is, however, not very great for a comparatively wide range of combustion on either side of the best rate.

Curves giving approximate draft pressures required for different rates of combustion when different kinds and sizes of fuel are hand fired are given in Fig. 213. The sizes referred to are explained in Tables XIII and XIV. Table XIII also shows the relative increase of ash content as the

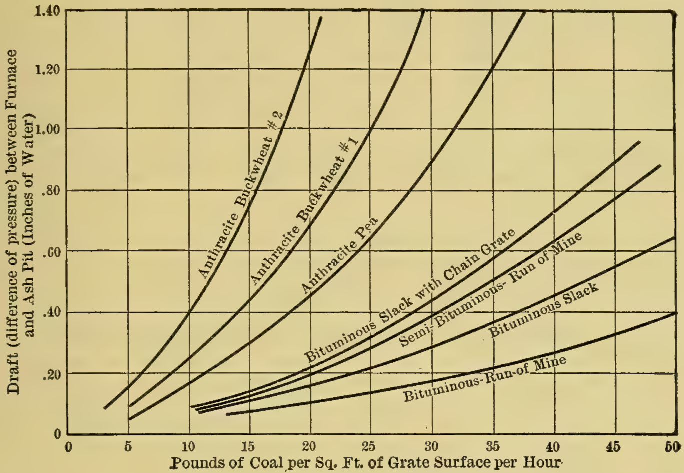


FIG. 213.—Draft Required for Different Rates of Combustion with Different Sizes and Kinds of Fuel.

size decreases, there being a tendency toward the concentration of the ash in the smaller sizes.

150. Strength and Safety of Boiler. Attention has already been called to the fact that the boiling vessel has to be designed with two different requirements in view: it must be mechanically strong to resist internal pressure and it must transmit the maximum amount of heat to the contained water.

Spherical and cylindrical surfaces with the pressure acting on the inside of the curve are best adapted to resist such pressures, as they already have the shape which the pressure would tend to give them. Boilers are, therefore,

TABLE XIII

Sizes of Anthracite Coal

(Sizes larger than pea coal generally too costly for power-plant use.)

Name.	Through Screen with Mesh. (Inclusive.)	Over Screen with Mesh. (Inclusive.)	Ash Content (Average).
Run of mine	unscreened	unscreened	
Broken	....	$2\frac{3}{4}$	
Egg	$2\frac{3}{4}$	2	6
Stove	2	$1\frac{1}{4}$	10
Chestnut	$1\frac{1}{4}$	$\frac{3}{4}$	13
Pea	$\frac{3}{4}$	$\frac{1}{2}$	15
Buckwheat No. 1	$\frac{1}{2}$	$\frac{1}{4}$	17
Buckwheat No. 2 or rice	$\frac{1}{4}$	$\frac{1}{8}$	18

TABLE XIV

Sizes of Bituminous Coal

(Considerable variation in commercial practice in [illegible]ing and sizing.)

Name.	Through Bars Spaced Apart. (Inches.)	Over Bars Spaced Apart. (Inches.)
Lump	....	$1\frac{1}{4}$
Nut	$1\frac{1}{4}$	$\frac{3}{4}$
Slack	$\frac{3}{4}$	

constructed as far as possible of vessels having only spherical and cylindrical surfaces.

Flat surfaces which are poorly adapted to resist such pressures as act within a boiler must often be used despite their weakness. When incorporated in a boiler they are invariably "stayed," that is, braced by being fastened to other surfaces by stay bolts and other forms of fastenings. Examples will be given later.

Most of the early designs of boilers and many of the modern types consist of large cylindrical vessels made by riveting together properly shaped steel plates. These shells are often traversed from end to end by flues or

tubes for carrying hot gases and generally have flat ends more or less perfectly braced by these tubes and by long tie rods and other braces. Such boilers when in operation

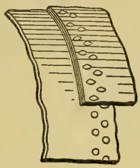


FIG. 214.—Lap Joint.

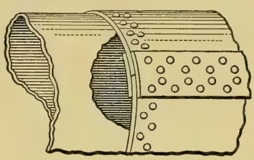


Fig. 215.—Butt Strap Joint.

are almost entirely filled with water and often hold many tons.

Boilers of these types have been responsible for many disastrous boiler explosions, and this fact has led inventors to the development of models which should be less dangerous. It seems practically impossible to develop a commercial boiler which cannot be made to explode to a certain extent

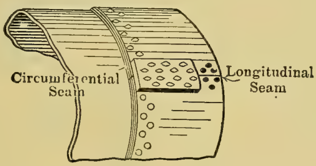


FIG. 216.—Riveted Plates of Boiler Shell.

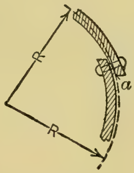


FIG. 217.

if sufficiently mistreated and mishandled, but much can be done to minimize the danger.

The great weakness of the older forms lies in the riveted joints, which can never be made as strong as the plates which they fasten together. Two types of joint are in use; they are known respectively as the **lap joint** and the **butt strap joint.** These are shown in Figs. 214, 215 and 216. So far as a circumferential seam, that is, one running around the cylinder as shown in Fig. 216, is concerned, the lap joint

is perfectly satisfactory and is universally used. With longitudinal seams, however, this is not the case. A lap joint throws the joined edges out of a true cylindrical surface as shown in Fig. 217, and when the vessel is subjected to pressure there will be a tendency for the plates to assume a cylindrical contour as nearly as possible. This causes local bending of the plates on each side of the lines of rivets, and the continued repetition of this action ultimately causes failure. The conditions are often made still worse by calking the joint on a line indicated by *a* in Fig. 217, that is, by hammering the metal at the inner surface of the edge of the outer plate into firmer contact with the outer surface of the inner plate for the purpose of making a tight joint.

The butt-strap joint can obviously be made so that the joined plates more nearly form a true cylindrical surface.

Other weaknesses of the older forms lie in the flat surfaces used; in constructions which render it possible for sediment to collect on heated surfaces and thus permit local overheating of the plate; and, above all, in the very large quantity of water contained.

The disastrous consequences of boiler explosions are generally due to the action of the hot water contained within the boiler and not to the steam contained at the time rupture occurs. The water within the boiler is under steam pressure and approximately at steam temperature. Removal of the pressure by rupture of the container would enable a great part of this water to flash suddenly into steam at the expense of its own heat, and this is exactly what occurs in the case of a boiler explosion. Local failure causes a sudden lowering of pressure, and this results in the formation of large volumes of steam which, blowing out through the initial fracture, tend to enlarge it, to move the boiler and surroundings, and, in general, to do all possible to further the rupture and make conditions worse.

From the preceding discussion the requirements for maximum safety can be deduced. They are:

1. The smallest convenient diameter of cylindrical vessels, so as to decrease the total load on joints for any given steam pressure.

2. The elimination of the greatest possible number of riveted joints and the use of butt-strap longitudinal joints on all large-diameter, cylindrical vessels.

3. The substitution of curved surfaces for all flat stayed surfaces.

4. So shaping the boiler that the required extent of heating surface may be obtained without enclosing a great volume to be filled with hot water when the boiler is steaming.

5. So shaping the boiler that such water as is contained therein will be divided up into small masses contained within separate vessels connected in such a way that rapid flow of all water toward one point of failure is impossible.

6. So shaping the boiler that no riveted joints shall be in the paths of flames and that no sediment can collect on metal immediately over flames or exposed to very hot gases.

7. So shaping the boiler that it shall be free to expand and contract with changes of temperature, with the least resultant strain on the different parts.

These various requirements are most nearly met in the different forms of water-tube boilers, some of which will be described in succeeding paragraphs.

Fig. 218.—Circulation in a Flask.

151. Circulation in Boilers. If a flask of water, such as that shown in Fig. 218, be heated in the manner indicated, the water will gradually acquire motion and follow paths such as those shown by the arrows in the illustration. The heated water will rise in the center of the mass and the cooler water will flow downward around the outer surface.

Such motion is called **circulation**. Rapid circulation within a boiler is very desirable, since it brings the maximum quantity of water in contact with the heating surfaces in a given time and hence tends to increase the amount of heat taken from those surfaces. It also tends to sweep along any bubbles of steam or gas formed on such surfaces and to carry away any sediment which may have collected, thus preventing overheating of the surfaces.

Circulation can be expedited by providing free and unrestricted paths for the water so as to guide it in the proper directions and by applying the most intense heat at the proper point along the path of the water. The temperature of the water which is subjected to the most intense

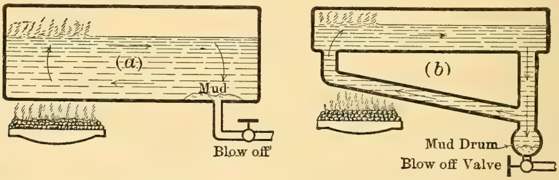


FIG. 219.—Elementary Types of Boilers.

heat is naturally raised and the water at that point becomes less dense than in other parts of the boiler. The formation of steam at such points also materially lessens the density. As a result of this lowering of density the heated water rises and the cooler water descends to take its place. The more rapid this exchange can be made, the more steam can be generated from a given amount of surface in a given time and hence, other things equal, the better the boiler.

The elements of two common forms of boiler are shown in Fig. 219, the arrows indicating the direction of the circulation and its effect upon the delivery of steam and of sediment.

152. Types of Boilers. In a book of this scope it would be impossible to describe all the types of boilers at present

in use. The more important varieties **have** therefore been chosen for description and illustration.

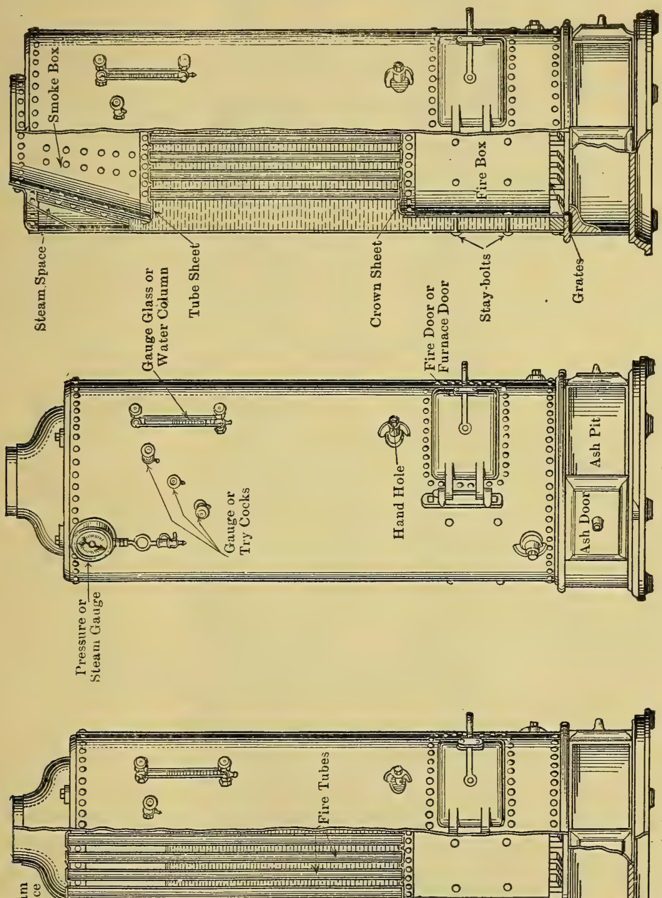


FIG. 220.—Upright or Vertical Fire-tube Boilers.

Two types of **internally fired, tubular boilers** more accurately described as internally fired, upright or vertical, fire-tube boilers are shown in Fig. 220. The furnace is

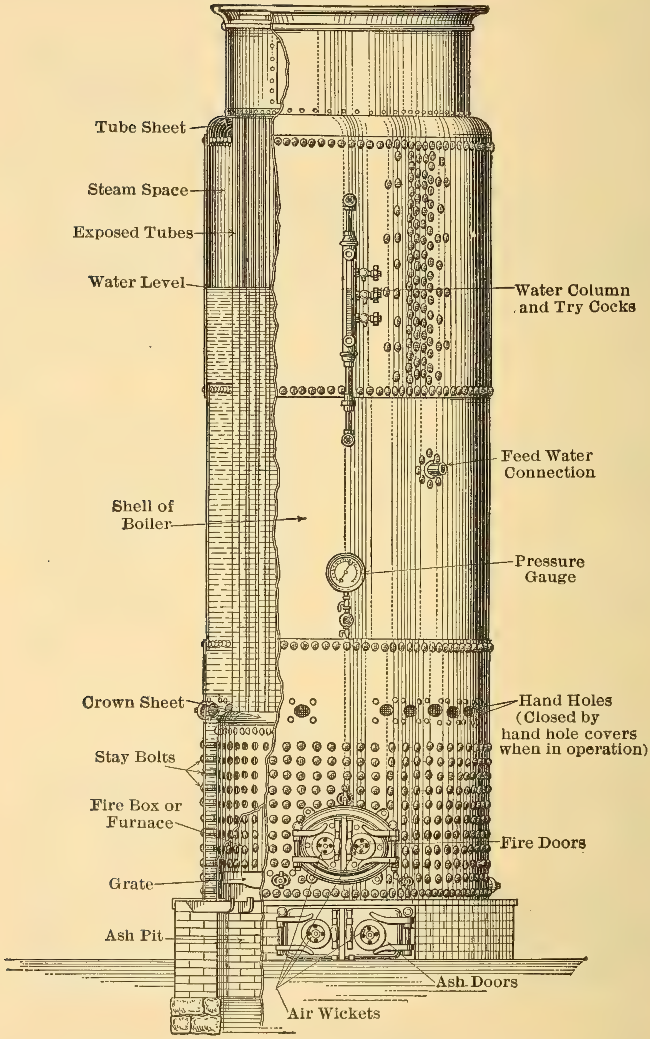


FIG. 221.—Large Internally Fired Tubular Boiler.

contained within the shell of the boiler and is almost completely surrounded with water. The heat radiated from the hot fuel is thus almost entirely received by the water of the boiler. The hot gases, rising from the fuel bed, pass upward through the tubes and, after giving up part of their heat to the surrounding metal, enter the smoke box and pass directly to the stack.

Boilers of the type shown in Fig. 220, (*a*) and (*b*) are called **exposed-tube** boilers, because the water level is carried below the tops of the tubes. The tubes, therefore, extend through the steam space and act as imperfect superheaters. Boilers of the type shown in Fig. 220, (*c*), in which the tubes do not enter the steam space, but are entirely covered by water, are called **submerged-tube** boilers.

Upright tubular boilers of the types shown in Fig. 220 are built by a number of manufacturers in sizes ranging from about 4 boiler horse-power to about 50 boiler horse-power. They are self contained, require no setting of any kind, and are shipped completely erected. Such boilers are very often mounted on trucks or skids and used to generate steam for small hoisting and other forms of contractors' engines. They are also used on steam fire engines.

The pressure carried in these small tubular boilers is generally under 100 lbs. per square inch, but they can be built for higher pressures if desired.

In Fig. 221 is shown a larger type of internally fired tubular boiler as made by the Bigelow Company for stationary use. These boilers are similar to those just described, but are made only in large sizes, in this case, in sizes ranging from 40 boiler horse-power to 200 boiler horse-power. The exposed tubes generally give a superheat of about 25° F.

These large upright boilers can be built to operate with a pressure as high as 200 lbs. per square inch and because of the small area covered by even the largest sizes, they are particularly adapted to locations in which floor space is limited.

The **locomotive type** of boiler is shown in Fig. 222. It is an internally fired, horizontal, tubular or fire-tube, boiler. Such boilers are seldom used for stationary purposes, but are universally used on steam locomotives and, in the smaller sizes, are often mounted on trucks or skids and used for semi-stationary purposes by contractors and others. Boilers of this type are built in sizes ranging from 10 boiler horse-power or less up to over 100 boiler horse-power for general power purposes, while those used on the largest locomotives generate over 2000 boiler horse-power.

The **Continental type** of boiler, named from the Continental Iron Works, is shown in Fig. 223. These boilers

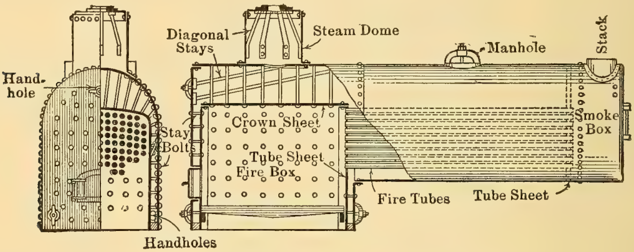


FIG. 222.—Locomotive Type of Boiler.

may be described as internally fired, return tubular, with semi-external combustion chamber, this chamber being outside of the boiler shell proper but being built as an integral part of the boiler and transportable therewith. Boilers of this type are built in sizes ranging from about 75 boiler horse-power to 300 or more.

The grates, furnace and ash spaces, and bridge wall are all carried within circular, corrugated flues, one flue being used in the smaller sizes and two in the larger. The corrugations serve the double purpose of strengthening the flue and of exposing added heating surface to fire and hot gases.

The steam pipe shown just below the steam connection at the top of the boiler is commonly used on boilers for the

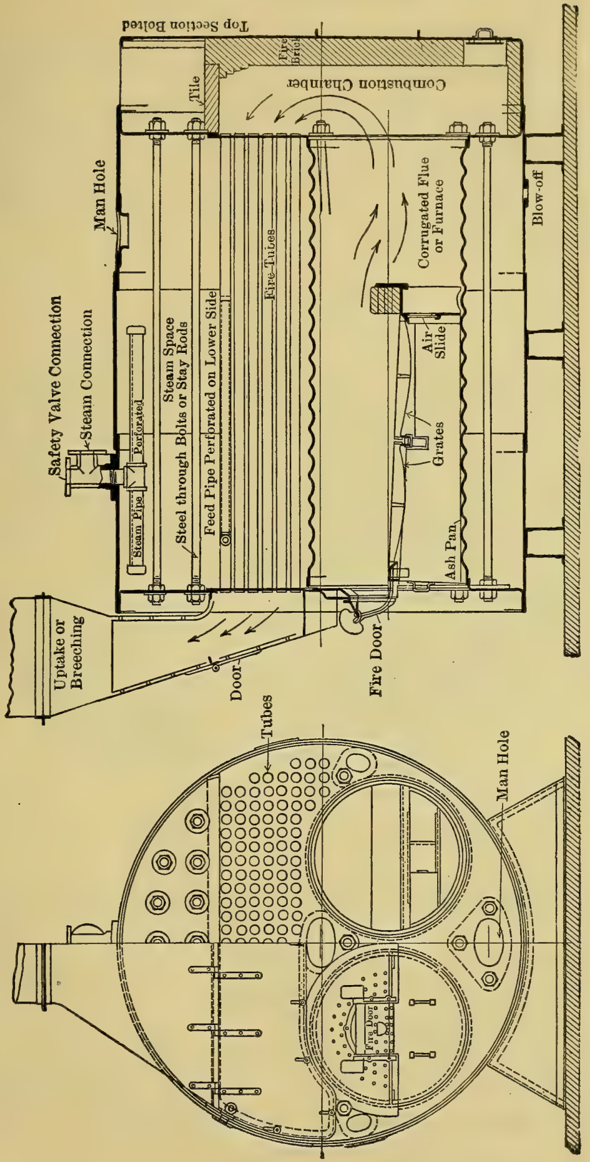


FIG. 223.—Continental Type Boiler.

purpose of preventing the escape of excessive quantities of moisture with the steam.

These boilers are very compact in shape and are short for their capacity, but they contain a great volume of water. They possess the advantages of having a large steam space and a very extended liberating surface over which the steam separates from the water.

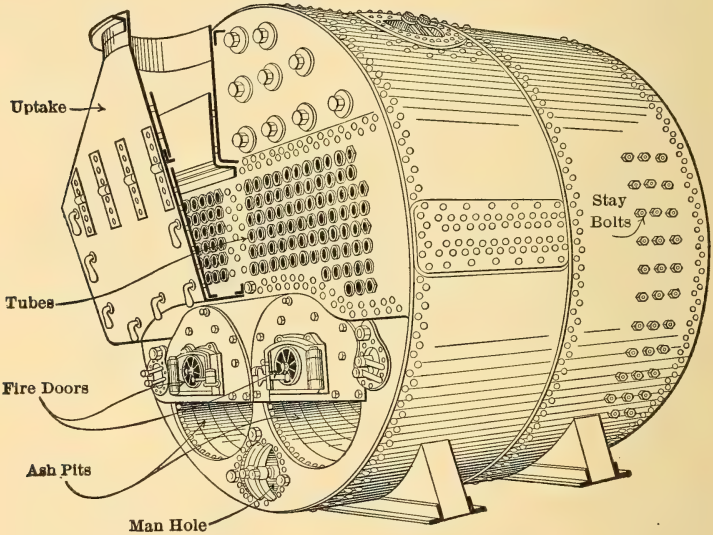


Fig. 224.—Scotch Marine Type Boiler.

The **Scotch marine type** of boiler is shown in Fig. 224. It has the same general construction as that just described excepting that the combustion chamber is entirely enclosed within the water space of the boiler. This chamber is built up of flat plates and is held against collapse by numerous stay bolts. Boilers of this type were until recently the standard for marine practice, but they are now being replaced in many instances by water-tube boilers of more recent design.

Scotch marine boilers are very economical in the use of fuel, are good steamers, and are absolutely self contained. They are built in numerous sizes, the smallest having shells with diameters of about 6 ft., while the largest diameter used is about 16 ft. The largest boilers have three and four corrugated furnaces.

Two types of **externally fired, return-tubular** (or "H.R.T.") **boilers** are shown in Figs. 225 and 226. The

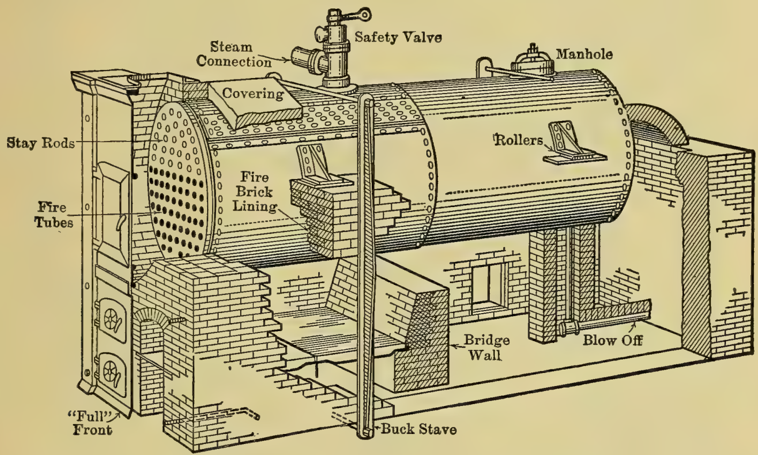


FIG. 225.—Horizontal Return-tubular Boiler with "**Full Flush Front.**"

only essential differences in these two types are in the forms of setting and in the methods of suspending the boilers. The shell is generally rigidly supported at the furnace end and arrangements made to allow for movement of the other end with changes of temperature.

These boilers can be built very cheaply and are therefore widely used when their limitations do not prevent. It has been found inadvisable to build them in sizes larger than 200 boiler horse-power or for pressures higher than 150 lbs. per square inch, and they are generally used in smaller

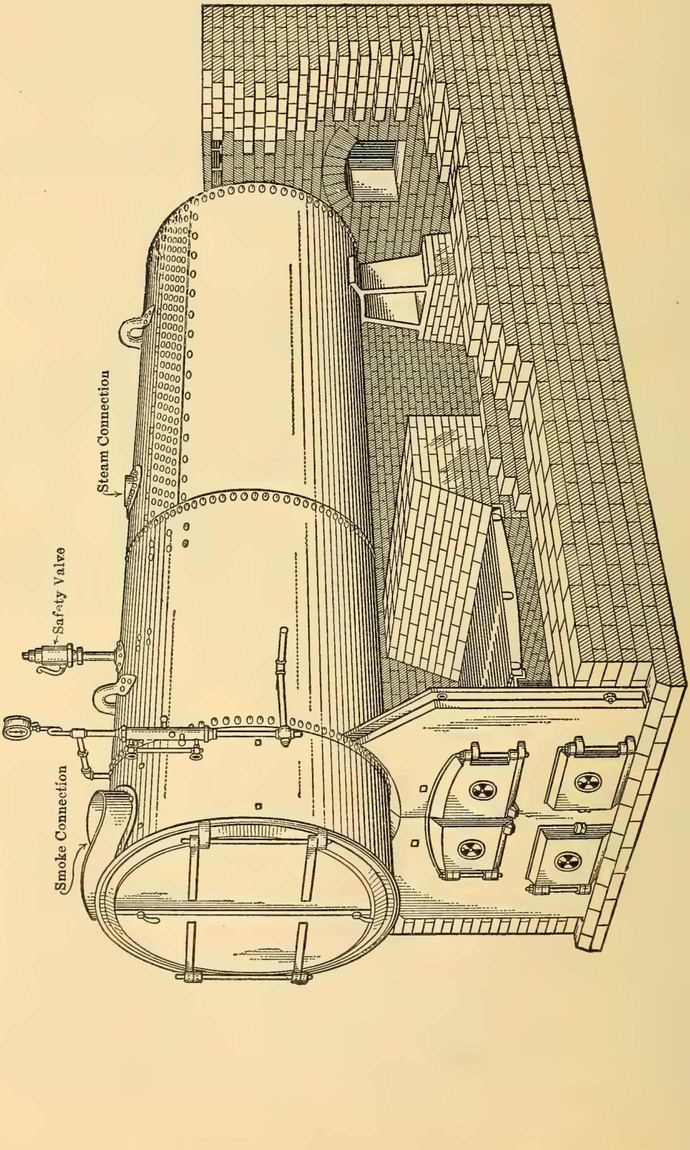


Fig. 226.—Horizontal Return Tubular Boiler with "Half Front."

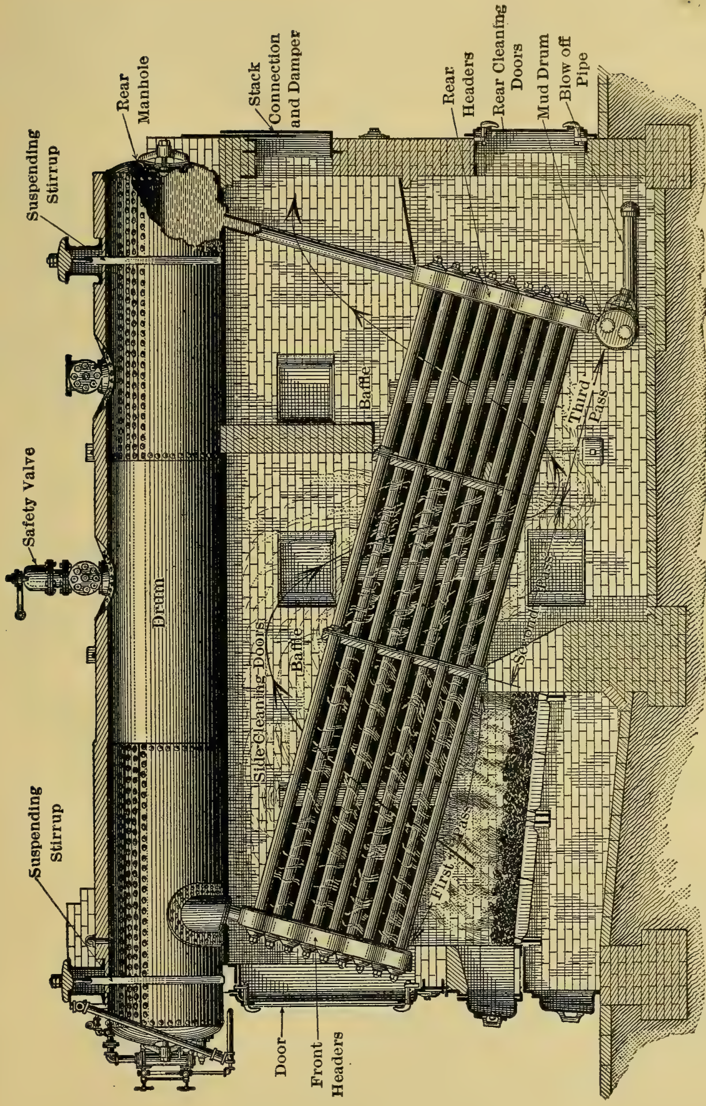


FIG. 227.—Babcock & Wilcox Water-tube Boiler.

sizes and with lower pressures. These limitations are set by permissible thickness of metal immediately above the fire, experience having shown that the plates deteriorate rapidly at this point if made too thick.

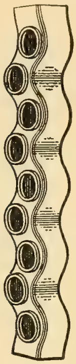


Fig. 228.—Forged Header for Babcock & Wilcox Boiler.

One form of **Babcock & Wilcox water-tube boiler** is shown in Fig. 227. This boiler is built up of sections consisting of several tubes joined at the ends by headers, and the sections are connected side by side at each end to a long horizontal drum. The ends of this drum are closed with "dished" heads, thus doing away with flat surfaces and the necessity for stays within the drum.

A detail of the forged header is shown in Fig. 228. It may be regarded as a long box of rectangular section with opposite walls pierced by circular holes, which has been so distorted as to give it a wavy shape. The distortion brings the holes into such positions that the tubes when expanded into these holes are "staggered," that is, do not lie one above the other.

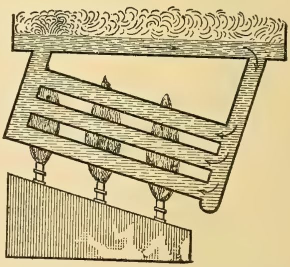


Fig. 229.—Elementary Babcock & Wilcox Boiler, Showing Circulation.

The general principle involved in the arrangement of these sections or elements and the resulting circulation are shown in Fig. 229. The location of the feed-water inlet and other details are shown in Fig. 230. It will be observed that the feed water enters in such a direction and position that it is readily picked up by the current of water circulating in the boiler, carried toward the rear and down the rear header. During this travel it is heated by contact with the hot water in

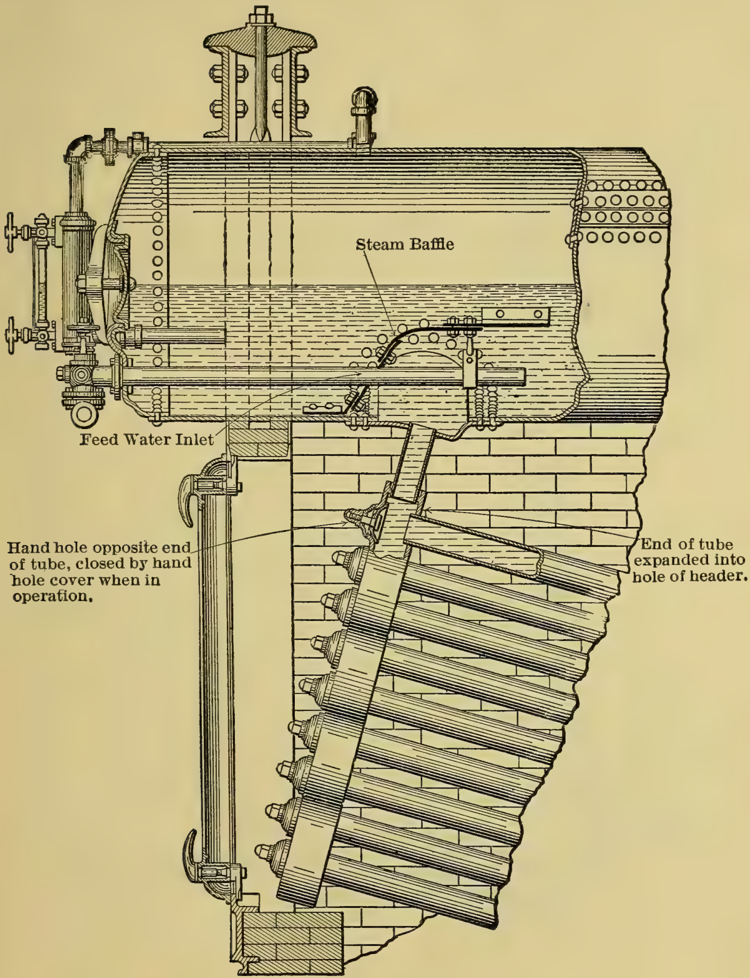


FIG. 230.—Details of Babcock & Wilcox Boiler Construction.

the boiler and most of its impurities are separated out and settle in the mud drum at the bottom of the rear header.

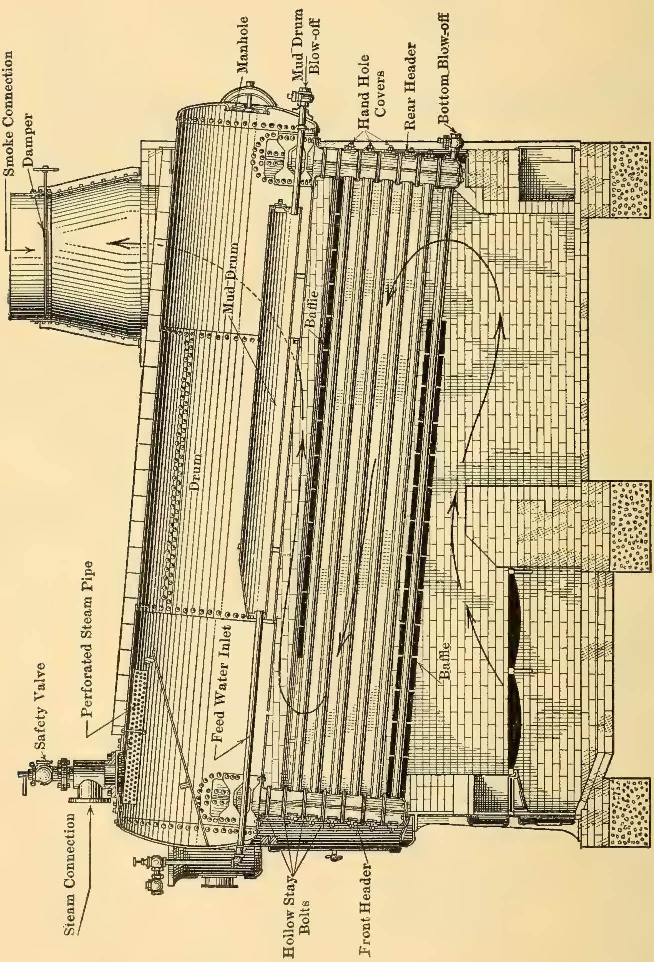


Fig. 231.—Longitudinal Section of Heine Boiler and Setting.

The boiler is suspended by stirrups from beams carried by the brickwork as shown in Fig. 227, the tube sections

simply hanging from the drum by the nipples at each end. The various parts of the structure are thus free to expand and contract independently as their temperatures change and are not bound in any way by the brick setting.

The steam is collected from a perforated steam pipe near the top of the steam space. The baffle shown in Fig. 230 prevents the steam which rises from the front header from carrying the water bodily into the steam space and makes the greater part of the water surface in the drum act as separating surface.

The scale which accumulates inside of the tube is removed by tools inserted through the hand holes in the front headers opposite the ends of the tubes. One of these hand holes and its cover are shown in section in Fig. 230. Soot and dust which accumulate on the outer surfaces of the tubes are blown off periodically by a steam jet, the necessary nozzle and hose being inserted through the tall and narrow side cleaning doors shown in Fig. 227 opposite each "pass."

A section of the **Heine water-tube boiler** is shown in Fig. 231. This boiler consists of a slightly inclined drum with dished heads, two sheet-steel headers and numerous tubes connecting these headers. The shape of the header is shown in Fig. 232, which indicates the positions occupied by the tubes and the way in which the header is joined to the drum.

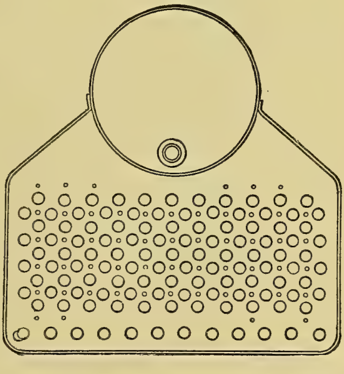


Fig. 232.—Front End Elevation, Heine Boiler.

The products of combustion are generally made to pass along the tubes by the longitudinal baffles shown, instead of across the tubes as in the boiler last described.

The mud drum in this type is located within the boiler and consists of a sheet-steel box supported a few inches above the bottom of the drum. The feed water enters at the front end of this drum and gradually spreads out as it is heated by the surrounding water. The greater part of the impurities settles to the bottom and is blown off periodically. The warmed water rises and flows out of an opening in the top of the box at the front end and there joins the circulation of the boiler, traveling toward the rear, down the rear header and up the tubes to the front header.

The interior of the tubes is cleaned of scale through hand holes just as in the last boiler. The external surfaces are freed of soot and dust by means of a steam jet which is introduced through the stay bolts in the headers, these bolts being made hollow for this purpose. Since it is not necessary to use doors in the side walls for cleaning in this type, Heine boilers are often set up in batteries of three or more, each interior side wall serving as the side wall of two settings. In the case of the boiler last described the necessity for side cleaning doors makes it impossible to join more than two boilers in this way.

The Heine boiler is supported by standing the front and rear headers upon the brickwork of the setting and it can therefore expand freely in all directions.

A section of the **Sterling water-tube boiler** is shown in Fig. 233. This boiler consists of three upper horizontal drums connected by short curved tubes and connected to a single lower horizontal drum by means of long tubes which are curved near the ends. The curves of all tubes are so made that the tubes enter the drum surfaces radially, thus giving a simple joint which is readily made tight by expanding the tube into the sheet.

The feed water is introduced into the upper rear drum, and is gradually heated and partly purified as it passes downward to the lower drum, in which the greater part of the material precipitated from the water is caught and

stored until blown off. From the lower drum the water is supposed to pass upward through the front bank of tubes, the steam formed passing to the central drum through the upper set of short curved tubes, and the water which is not evaporated passing to the central drum through the

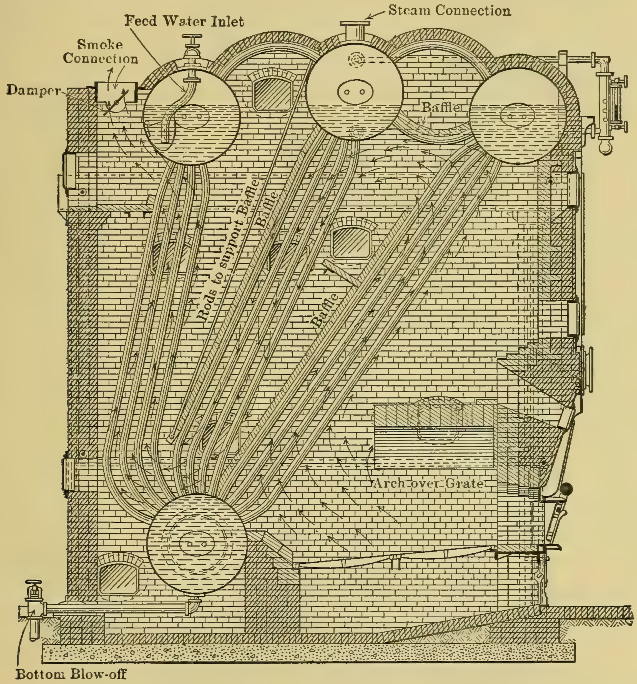


FIG. 233.—Section of Sterling Boiler.

lower set of curved tubes. This water passes from the upper central drum to the lower and returns through the front bank of tubes. Any steam formed in the rear bank of tubes or in the rear drum passes to the central drum through the short curved tubes connecting the steam spaces.

The entire boiling vessel is hung from a frame of structural steel by means of the upper drums, so that the lower

drum hangs practically free on the tubes. Independent expansion of all the members is insured by this method of suspension and by the curvature of the tubes, which permits each one of them to bend to the extent necessary to equalize any strains caused by changing temperatures.

The interiors of the tubes are cleaned by means of tools lowered from inside the upper drums and the exterior surfaces are blown off by steam jets introduced through doors in the brickwork of the setting.

The **Wickes vertical water-tube boiler** is shown in section in Fig. 234. It consists of an upper and lower circular drum, connected by straight tubes expanded into the lower and upper heads of the drums respectively. A vertical baffle placed in the center of the bank of tubes gives an upward path to the products of combustion when passing over the front tubes and a downward path when passing over the rear tubes.

The feed water is generally introduced at the rear of the upper drum, the circulation being downward in the rear tubes and upward in the front tubes.

The interior surfaces of the tubes are cleaned by tools lowered into them by a man standing within the upper drum, which is made high enough to make this possible. The external surfaces are cleaned by steam jets inserted through doors in the brickwork.

The entire boiler is supported on brackets riveted to the lower or mud drum and is free to expand in all directions, the brickwork simply enclosing but not confining it.

153. Boiler Rating. Practically all apparatus which is connected with the development of power is given a *horse-power* rating. In some cases such a method of rating is convenient and simple, in others it is inconvenient, irrational and complicated. The term horse-power, when used as a measure of work or power, means very definitely the equivalent of 33,000 ft.-lbs. per minute. When, however, a certain number of horse-power is used as the rating of a

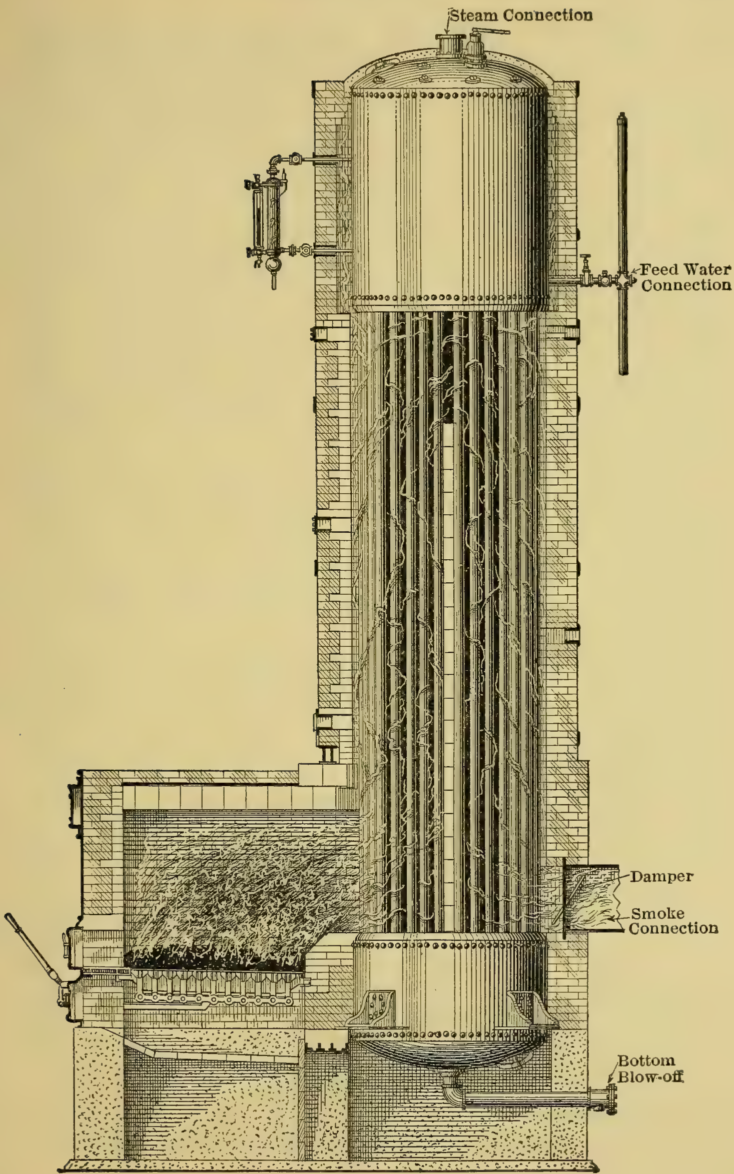


FIG. 234.—Wickes Vertical Boiler.

particular piece of apparatus, it generally means that that piece of apparatus, when working at about its best efficiency, can do what is necessary to make available the stated number of horse-power in the plant of which it forms a part.

Thus a boiler rated at a certain horse-power was originally supposed to be able to supply the amount of steam required by an average engine developing that quantity of power and to do this when working at its best efficiency. The water rates of engines are, however, so different that there is no real connection between boiler horse-power and engine horse-power, and it is best to consider the boiler horse-power as a perfectly arbitrary unit defined in a certain way.

The American Society of Mechanical Engineers has defined the **boiler horse-power** as the *equivalent of the evaporation of* 34.5 *lbs. of water per hour from and at* 212° *F.* This means the conversion per hour of 34.5 lbs. of water at 212° F. into steam at the same temperature and therefore at atmospheric pressure.

Each pound of steam generated under these conditions requires the expenditure of the latent heat of vaporization at atmospheric pressure, which is equal to 970.4 B.t.u. according to the latest steam tables. The older tables gave 965.7. This quantity of heat is known as a Unit of Evaporation and is abbreviated U.E. The boiler horse-power is, therefore, the equivalent of 34.5 U.E. per hour or $34.5 \times 970.4 = 33{,}479$ B.t.u. per hour.

As practically no power-plant boilers receive their feed water at a temperature of 212° F. and convert it into steam at the same temperature, it is necessary to convert the weight actually evaporated to what it would have been from and at 212° F. and then to divide this figure by 34.5 in order to find the boiler horse-power developed.

The number of pounds which would have been evaporated from and at 212° F. if the same amount of heat had been

transmitted is known as the **equivalent evaporation**, or as the *equivalent weight of water evaporated into dry steam from and at* 212° *F.*

The method of obtaining the equivalent evaporation has been defined by the American Society of Mechanical Engineers. The heat given to each pound of dry saturated steam produced is to be determined; this is to be multiplied by the total weight of dry saturated steam generated per hour, and the product is to be divided by the latent heat of vaporization at 212° F. Thus, for a boiler receiving its feed water at some temperature t_f above 32° F., the water contains a quantity of heat equal to q_f B.t.u. per pound, q_f being found in the steam table opposite the temperature t_f. Each pound of dry saturated steam leaving the boiler carries with it an amount of heat equal to λ for the existing temperature. The heat supplied each pound in the boiler must therefore be $\lambda - q_f$ and, for W pounds per hour, the heat supplied would be $W(\lambda - q_f)$. The equivalent evaporation is then given by

$$\text{Equiv. evap.} = W\left(\frac{\lambda - q_f}{970.4}\right) \text{ lbs. per hour.} \quad . \quad (101)$$

This expression may be regarded as consisting of two factors, the weight of dry steam generated per hour, and a fraction which will always have the same value for a given combination of pressure and feed-water temperature. This fraction is called the factor of evaporation, and it is customary to tabulate the various values of the factor of evaporation for different common combinations of pressure and feed-water temperature.

It should be noted that the equivalent evaporation as defined above gives the boiler no credit for heat given to water which leaves the boiler as water, nor does it give credit for any superheating. The former may be justified by saying that the boiler, as a commercial piece of apparatus, is not intended to supply hot water; but many commercial

boilers are expected to supply superheated steam and should be given credit for heat used in that way.

Returning now to the boiler horse-power, its value can obviously be found for any given boiler by dividing the equivalent evaporation per hour by the number 34.5.

Boilers are supposed to be so rated that they will develop their rated horse-power when operating at about their best efficiency and will do it with moderate draft and reasonably good firing with average fuel. Experience has shown that for most boilers the best efficiency is obtained when an equivalent evaporation of from 3 to 3.5 lbs. of water occurs per square foot of heating surface. The heating surface is generally taken as the total surface in contact with hot gases excepting in the case of tubes. The outer surfaces of tubes are generally counted even if they be in contact with the water. An equivalent evaporation of 3 to 3.5 lbs. per square foot would call for a heating surface of from 12 to 10 sq.ft. per boiler horse-power.

Most water-tube boilers are given 10 sq.ft. of heating surface per rated boiler horse-power, and most return-tubular boilers are supplied with 11 to 12 sq.ft. Scotch marine boilers are generally designed on a basis of about 8 sq.ft. per rated boiler horse-power.

The quantity of water which can be evaporated per square foot seems to depend to a great extent upon the rate at which hot gases can be passed over the heating surface, and experiments have shown that from five to eight times the ordinary rates of evaporation can be attained if sufficient fuel can be burned. As the rate of evaporation per square foot is increased above the commonly accepted value, the efficiency decreases, but the decrease is generally small for a considerable increase in rate of evaporation. Most power-plant boilers can give from 150 to 200 per cent of their normal rating, and some are now installed to operate for long periods at about 200 per cent of what would be considered a normal rating.

154. Boiler Efficiencies. There are a great many possible efficiencies which may be considered in connection with boiler tests. The two most commonly used are defined by the A.S.M.E., and are:

1. Efficiency of the boiler

$$= \frac{\text{Heat absorbed per pound of combustible burned}}{\text{Calorific value of 1 lb. of combustible}}.$$

2. Efficiency of boiler and grate

$$= \frac{\text{Heat absorbed per pound fuel}}{\text{Calorific value of 1 lb. of fuel}}.$$

The names used are not very well chosen, and it is better to call the first the *efficiency based on combustible* and the second the *efficiency based on coal.* The weight of combustible burned is calculated by subtracting from the coal fired the total weight of moisture and the total weight of refuse in the ash pit.

The heat absorbed is by definition the heat absorbed by the dry steam made by the boiler, but it seems probable that this will also be modified in the near future as suggested in a preceding paragraph.

It is also possible to determine the efficiency of the grate, of the furnace, and of the boiling vessel, and this is sometimes done.

The best commercial operating values for the efficiency of the boiler as a whole, that is, the boiler and grate on the basis of total fuel fired, are about 75 per cent for good qualities of coal and 80 per cent for oil, but such values are generally obtained only in well-equipped plants operating on comparatively constant loads. Average commercial values generally range from 60 to 70 per cent on a yearly basis in well-equipped plants which are carefully operated, and many boiler plants are operated at an efficiency of 50 per cent and less.

The pounds of water evaporated per pound of coal

fired generally ranges between 6 and 10, and the equivalent evaporation per pound of combustible burned will generally fall between 8 and 12 pounds.

155. Effects of Soot and Scale. The flue gases in real boilers are seldom clean mixtures of the products of combustion and nitrogen, as theory would indicate. They always contain more or less soot and unburned hydrocarbons, as well as some finely powdered ash and fuel. With strong draught, very large particles of ash and fuel may be carried by the flue gases.

These materials are partly carried up the stack by the gases and partly deposited on the heating surfaces of the boiler. Such deposits decrease the conductivity of the heating surfaces, and if the deposits are heavy the loss may be very great. The results of one investigation on the effect of soot are given in Table XV, the values being taken from an article published in the Proceedings of the Institute of Marine Engineers for the year 1908. These values are probably too high, particularly for the thicker deposits, but they serve to bring out the fact that a very appreciable loss does occur from the presence of such deposits.

TABLE XV

EFFECT OF SOOT DEPOSITS ON BOILER HEATING SURFACES

Thickness of Deposit in Inches.	Loss of Conductivity in Per cent.
0	0.0
$\frac{1}{32}$	9.5
$\frac{1}{16}$	26.2
$\frac{1}{8}$	45.2
$\frac{3}{16}$	69.0

The effect of soot deposits in decreasing the efficiency of boilers was used for a long time as a basis for argument in favor of certain types of boilers in which the heating surfaces were so shaped and located that such deposits

formed to a minimum degree and against other types less favorably designed from this point of view. Practically, however, the removal of such deposits by means of steam jets applied at regular intervals is so simple that this consideration need be given little weight in the selection of a boiler. Provision should always be made, however, for the easy use of the jets for cleaning purposes.

156. Scale. Practically all water available for boiler feed contains various salts in solution and it often contains solid matter in suspension as well. This material is all deposited within the boiler as the water is heated and converted into steam. There is thus a gradual collection within the boiler of all the solid material brought in by the water.

In well-designed boilers the greater part of such deposits is carried to a part of the boiler in which the metallic surfaces are not exposed to high temperature gases, as, for instance, the mud drums in water-tube boilers. It can then be drawn off periodically in the form of a thin mud suspended in water. In practically all boilers, however, some of the solid material will be carried to the heating surfaces exposed to high temperature gases and deposited there. Under the action of heat, the mud-like material gradually changes until, in many instances, it forms a very hard, stone-like coating on the heating surface. This is known as boiler scale.

Such deposits may cause two kinds of trouble: They may decrease the conductivity of the heating surfaces and thus decrease the efficiency of the boiler; and, because of their location on the water side of the metal, they may permit the hot gases to overheat that metal, thus weakening it. Such overheated metal often "bags" under the high internal pressure and may eventually give way with disastrous results. The mechanical structure of the scale seems to be the determining factor; scales which are easily penetrated by water have little effect, while those which are very dense and non-permeable may cause serious trouble.

Boilers should be blown down periodically to keep them as free as possible of scale-forming material, and they should be so constructed that scale which has been formed can be removed easily. Very efficient tools have been developed for removing scale from the interior and exterior surfaces of tubes, so that boilers using tubular heating surfaces are readily cleaned of scale.

157. Scale Prevention. Much of the solid material carried by water is deposited when the water is heated to a temperature of from 150° to 200° F., so that heating feed water before it is admitted to the boiler is at least a partial preventive in most cases.

Nearly all of the salts which are soluble in hot water and therefore are not deposited when the feed is heated, can be made to form insoluble compounds by the addition of comparatively cheap chemicals. By the addition of such chemicals in the feed-water heaters, or in other apparatus specially designed for that purpose, the greater part of the solid content of the water can be precipitated before it is admitted to the boiler.

There are a great many "boiler compounds" on the market which are intended to be mixed with the water as it is fed to the boiler and are supposed to prevent the formation of scale on the heating surfaces. All they can possibly do is to change the chemical composition of the solids; they cannot prevent the deposit of these solids within the boiler. They are therefore, at best, only an imperfect remedy.

158. Superheaters. Many boiler plants are now arranged to supply steam superheated 25 to 200 degrees Fahr. It was shown in an earlier chapter that the use of superheated steam greatly improves the economy of reciprocating engines and turbines, and there are also other advantages which accrue from its use.

Superheaters are of two kinds—**separately fired** and **built-in superheaters**. The separately fired superheaters are

enclosed in a brick setting fitted with grate and furnace similar to that of an ordinary boiler. The built-in superheaters are installed within the boiler setting so that the products of combustion pass over them in flowing through the boiler.

In either type the steam passes through the superheater on its way from the boilers to the engines. In the case of separately fired superheaters, the temperature of the superheated steam is controlled by regulation of the fire on the grate of the superheater, but in the built-in type regulation in this way is practically impossible, as the fire under the boiler must be controlled to suit the demand for steam. The control of such superheaters is therefore effected either by locating them in such a position that the natural variation in the temperature of the gases reaching them gives an approximate regulation, or they are installed in a separate chamber and hot gases passed over them in such proportions as necessary to give the required temperature.

The Babcock & Wilcox superheater as applied to the boiler of the same make is shown in Fig. 235. The steam collected in the dry pipe within the drum passes downward to the upper manifold of the superheater and from there it flows through the U-shaped tubes into the lower manifold. From the lower manifold it flows through the superheater stop valve to the engine or turbine.

The superheater is so located that the hot gases pass over it between the first and second passes and there is no way of shutting off these gases. Provision, as shown in the illustration, is therefore made for flooding the superheater during starting, or when superheated steam is not desired. When flooded it becomes heating surface similar to that of the tubes below, the steam made passing into the drum through the dry pipe.

The Heine superheater as applied to a Heine boiler is shown in Fig. 236. It consists of a sheet-metal header or box into which U-shaped tubes are expanded. The steam

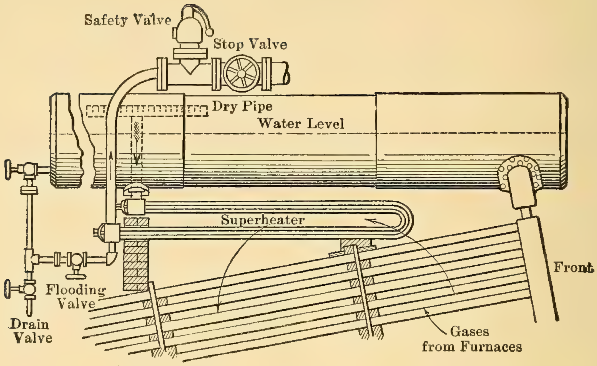


FIG. 235.

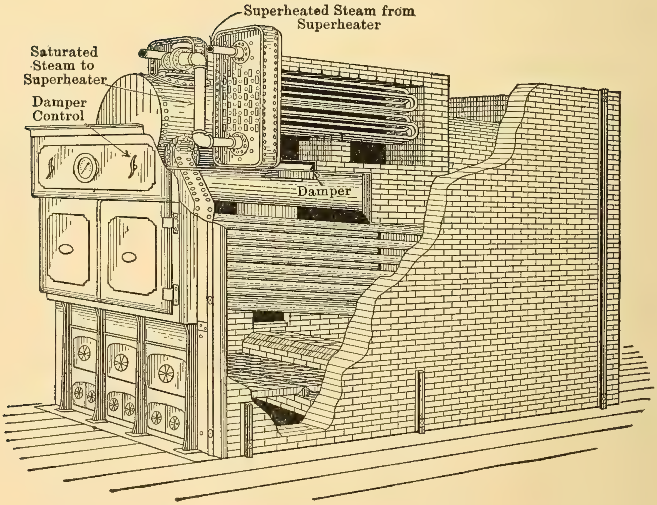


FIG. 236.—Heine Superheater.

enters the bottom of the header and is guided by diaphragms in such a way that it passes through the lower set of U-tubes, returns to the header, passes through the upper set of tubes, and then leaves the superheater at the top.

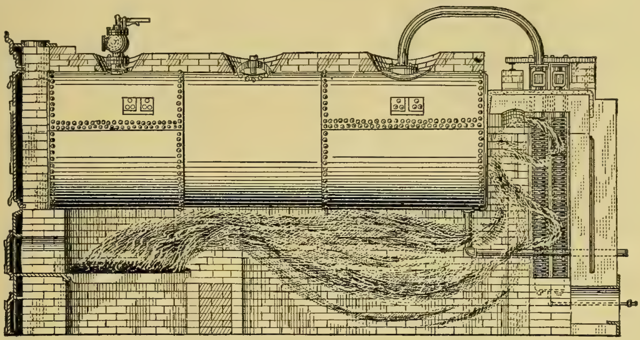


FIG. 237.—H.R.T. Boiler and Foster Superheater.

This apparatus is installed in a brick chamber built into the boiler setting and connected with the furnace by a flue (not shown) in the brick side wall. A damper controls the flow of hot gases to this chamber and the degree of superheat is controlled by the position of this damper.

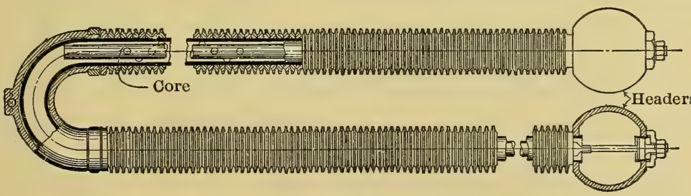


FIG. 238.—Element of Foster Superheater.

The Foster superheater is shown installed in the setting of an H.R.T. boiler in Fig. 237 and the details of the construction of one element are shown in Fig. 238. The core is used to spread the steam in a thin stream, thus bringing it into better contact with the heating surface. The fins

on the exterior of the element are used for the purpose of getting a more extended metallic surface in contact with the hot gases.

159. Draft Apparatus. Attention was called in a preceding paragraph to the fact that there must be a difference of pressure between the spaces below and above the fuel bed in order to cause the necessary air to flow through the bed. This difference of pressure is called the **draft.**

As a matter of fact, a slight difference of pressure is required to cause the flow of gases through any part of the boiler and the drop in pressure through the fuel bed is only part of the total draft required.

The draft may be created in two distinctly different ways. It may be caused by a chimney or stack, and is then known as **natural draft**, or it may be produced by fans or blowers, in which case it is called **mechanical draft.**

(*a*) **Chimneys or Stacks.** Stacks are practically always used in small plants because of the simplicity resulting from their use and because the interest on the investment compares favorably with interest on investment plus cost of operation for mechanical draft. In large plants fitted with some types of mechanical stokers, or where fuel is to be burned at a high rate, or where the flue gases are to be used for heating feed water, mechanical draft is generally installed. A stack of some sort is necessary even though mechanical draft be used, because the products of combustion must be discharged at a sufficient elevation to prevent their being a public nuisance.

A chimney serves to carry away the hot products of combustion and when in operation is filled with a column of gases with higher average temperature than that of the surrounding air. As a result the density of gases within the stack is less than the density of the outer air and the gas pressure at the bottom of the structure is less inside the stack than it is outside. If an opening is made at this point, the external air will therefore flow in. By arranging

the apparatus as shown in Fig. 239, the temperature of the air flowing into the bottom of the stack is raised as it passes through the furnace and the flow is thus made continuous.

The height of the chimney determines the draft created by it with flue gases of a given temperature, and, with any given height, the area determines the quantity of gas which

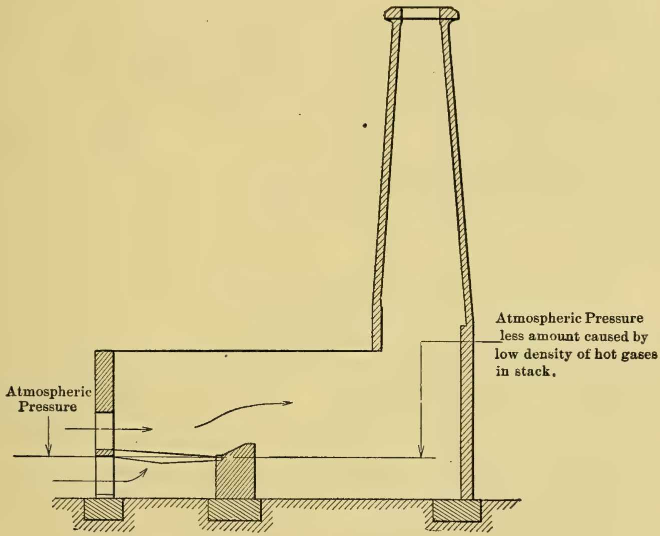


FIG. 239.—Diagrammatic Arrangement of Stack.

can be carried off in a given time. The proportions of chimneys can be determined from rational formulas based on theoretical considerations, but it is necessary to assume values for a number of constants and a proper choice depends largely upon experience.

As a result, all but the more important chimneys are generally designed on an empirical basis and many formulas have been developed for this purpose. One of the most common methods of design is to choose the height in accord-

ance with the values given in Table XVI, and then to determine the sectional area according to an empirical assumption or formula.

TABLE XVI

COMMON HEIGHTS OF CHIMNEYS

(Applicable to plants smaller than about 700 H.P. Larger installations should have stacks of from 150 to 175 feet in height unless local conditions call for greater height.)

Character of Fuel.	Height above Grate in Feet.
Free-burning bituminous................	80
Anthracite, medium and large sizes.......	100
Slow-burning bituminous................	120
Anthracite, pea size....................	130
Anthracite, buckwheat sizes............	150

Thus, some designers simply assume the sectional area at the top of the stack equal to about one-ninth of the grate area for anthracite coal and equal to about one-seventh of the grate area for bituminous coal. Others use a formula developed by William Kent, which is based upon the assumption that the stack should be large enough to carry away all the gases resulting from the combustion of 5 lbs. of coal per rated boiler horse-power per hour. This formula gives the boiler horse-power which the stack can serve and is

$$H.P. = 3.33(A - 0.6\sqrt{A})\sqrt{H}, \quad . \quad . \quad . \quad (102)$$

in which

H.P. = Rated boiler horse-power;

A = Internal sectional area in feet of circular or square chimney;

H = Height above grate in feet.

(*b*) **Mechanical Draft.** Fans can be so used as to force air into the ash pit, that is, to raise the pressure on the

entering side of the fire. In such cases the equipment is said to give forced draft. Or fans may be installed at the discharge end of the flues and may "draw" the gases through the boiler by lowering the pressure within to a value below that of the external atmosphere. Such an installation is said to give induced draft.

Forced draft suffers from the disadvantage that the pressure within the furnace is greater than atmospheric and hot gases may therefore be blown out when the fire door is opened. On the other hand, the fan handles only cool air instead of hot products of combustion as in the case of induced draft and its useful life is therefore much longer. Forced draft is much more common than induced draft.

Several arrangements giving *balanced* draft have been developed. With such apparatus a pressure equal to atmospheric is maintained above the fuel bed and no hot gases are blown out through the firing door.

PROBLEMS

1. The equivalent evaporation of a boiler during a certain test was 3450 lbs. per hour. What boiler horse-power was developed?

2. A water-tube boiler with 5000 sq.ft. of heating surface and rated in the ordinary way gave an equivalent evaporation of 25,875 lbs. per hour. At what per cent of rating was the boiler operating?

3. A certain boiler produced 3500 lbs. of dry steam in one hour from feed water at a temperature of 50° F. The steam pressure was 200 lbs. per square inch gauge. What was the equivalent evaporation?

4. A boiler receiving water at a temperature of 250° F. converts it into superheated steam at a pressure of 210 lbs. per square inch gauge and a temperature of 580° F. The boiler produces 26,000 lbs. of steam per hour. What is the equivalent evaporation if the boiler is given credit for all the heat given the material passing through it? What boiler horse-power is developed?

5. A boiler produces 7.5 lbs. of dry steam per pound of coal fired. The feed-water temperature is 80° F. and the steam pres-

sure is 125 lbs. per square inch absolute. What is the equivalent evaporation per pound of coal?

6. A boiler is supplied with coal which has a calorific value of 13,520 B.t.u. per pound. It produces 8 lbs. of dry saturated steam at a pressure of 150 lbs. per square inch gauge per pound of coal. The feed-water temperature is 70° F. What is the efficiency of the outfit?

CHAPTER XVIII

RECOVERY OF WASTE HEAT

160. Waste Heat in Steam Plant. There are two great heat wastes in the steam plant—the waste in the hot gases going up the stack and the waste in exhaust steam. The magnitude of the stack loss can best be appreciated by determining an approximate value for assumed conditions. For this purpose assume the fuel to be pure carbon, the excess coefficient 1.5, average atmospheric temperature 60° F., average stack temperature 600° F., and no moisture in the air. The specific heat of the flue gases may be taken as constant and equal to 0.24.

With an excess coefficient of 1.5, the total weight of flue gas per pound of carbon burned would be about 18.4 lbs. and the heat carried up the stack figured above room temperature would be

$$\text{Stack loss} = 18.4 \times 0.24\ (600 - 60).$$
$$= 2380 \text{ B.t.u. per pound of C burned (approx.)}$$

With a calorific value of 14,600 B.t.u. per pound of carbon this loss would be equivalent to a little over 16 per cent of the total heat in the fuel.

It would be more correct to use the temperature of the steam in the boiler instead of room temperature, because the lowest temperature theoretically attainable by gases passing through a boiler would be equal to that of the steam and water on the other side of the heating surface. Under ordinary conditions of operation, this method of figuring would give a theoretically avoidable stack loss equal to about 50 per cent of the figure obtained above.

The magnitude of the exhaust loss can be similarly approximated. Assume for this purpose an engine receiving dry saturated steam at 115 lbs. absolute per square inch and exhausting it with a quality of 90 per cent at a pressure of 15 lbs. absolute per square inch.

The heat above 32° in the entering steam is 1188.8 B.t.u. per pound and the heat exhausted per pound is 1053.7. The heat in the exhaust represents therefore about 89 per cent of all the heat supplied when calculations are made above a temperature of 32° F. If a feed-water temperature of 60° be assumed and heat quantities be figured above that datum the results are practically the same.

There are always numerous pieces of auxiliary apparatus in steam plants such as boiler-feed pumps, circulating pumps, vacuum pumps, etc. These are often steam driven and are generally very uneconomical in the use of heat, so that they throw away in their exhaust steam large quantities of heat originally transferred from fuel to water and steam in the boiler.

161. Utilization of Exhaust for Heating Buildings. It often happens that steam-power plants are located within or in the neighborhood of buildings requiring artificial heat during part of the year. In such cases the exhaust steam from main and auxiliary engines can generally be advantageously used for this purpose. Under particularly favorable circumstances, the weight of steam required by the plant may equal approximately that required for heating, and the greater part of the exhaust could then be turned directly into the heating system.

The engines in plants of this character may be regarded as reducing valves for the heating system, receiving steam at high pressure and reducing the pressure to the value best adapted to the heating system installed. If the comparatively small losses arising from radiation from the engine, from friction and from the presence of hot water in the exhaust be neglected, all heat received by the engine

and not turned into useful mechanical energy is made use of in the heating system. The engine may therefore be very uneconomical in the use of steam and still not cause a waste of fuel, provided always that the heating system can absorb all heat exhausted.

Since the demands of a heating system vary from day to day and since there is generally no demand for heat during several months of each year, it follows that a high degree of skill is necessary in choosing the character of the apparatus installed. A compromise is generally made between the cheap and uneconomical engine allowable during the coldest months and the more expensive and more efficient engine desirable when no heating is to be done.

There are other cases of somewhat similar character. In many industries use can be made of exhaust steam for the heating of evaporating pans, dye vats, kilns and other apparatus. Steam plants of an uneconomical character may be very economical financially in connection with such industries if all or nearly all of the heat in the exhaust can be utilized industrially.

162. Feed-water Heating. An examination of the steam table will show that the total heat above 32° F. per pound of saturated steam varies between 1180 and 1200 B.t.u. for such pressures as are commonly used in boilers. The average temperature of water as it occurs on the surface of the earth is probably somewhere in the neighborhood of 60°, so that the heat above 32° per pound would roughly average 27 B.t.u. A boiler receiving water at 60° and converting it into steam at any of the ordinary pressures must therefore supply over 1100 B.t.u. per pound of water.

This immediately suggests a use for heat in exhaust steam. Steam exhausted into very low vacuums has a temperature only 10° to 30° higher than the assumed average natural feed temperature, but steam exhausted at atmospheric pressure has a temperature of 212° F. and could therefore impart large quantities of heat to water at 60° F.

Since the boiler must supply over 1100 B.t.u. per pound of steam made, raising the feed temperature about 11° or 12° should effect a saving of about 1 per cent in fuel consumption. By raising the temperature from 60° to 212° there should therefore result a saving of approximately 13 to 14 per cent.

Other advantages which would accrue from this preliminary heating of the feed water would be (1) the deposit, outside of the boiler, of a large amount of the solid matter carried by the water, (2) the use of fewer or smaller boilers, and (3) the reduction of the strains which occur in the metal of some designs when very cold feed water is used.

Exhaust steam feed-water heaters are divided into two types, *open* and *closed heaters*. In **open heaters** the steam and feed water are brought into intimate contact in the form of jets, sheets and sprays within a vessel of appropriate size and shape. They are often called contact heaters. When the exhaust steam comes from reciprocating engines it always carries in suspension some of the oil used for lubricating the engine cylinders. If allowed to enter the heater, this oil would mix with the feed water and eventually reach the boilers, where it might cause serious damage by depositing upon heating surfaces exposed to the fire or to very hot gases. Such heaters are therefore always fitted with oil or grease extractors when used with reciprocating units. When receiving the exhaust from turbines, oil extractors are not necessary, as no lubricant is used within the steam spaces of such units.

Closed heaters consist of tubes or coils enclosed within a metal vessel. One medium passes through the tubes and the other over their outer surfaces. Such heaters are therefore often called non-contact heaters.

As oil is a poor conductor of heat, the exhaust steam from reciprocating units should be passed through an oil extractor before entering a closed heater in order that the heating surfaces may be used to the best advantage.

Exhaust steam feed-water heaters are often divided into *primary* and *secondary heaters*. This distinction has nothing to do with structure, being based entirely on position and temperature. Thus there may be available exhaust steam at a pressure below atmospheric, as from condensing main units, and exhaust steam at atmospheric pressure from non-condensing auxiliaries. The lower pressure steam could be used to heat the feed water in a primary heater and the higher pressure steam could then raise its temperature still further in a second or secondary heater.

The other great waste, that in the stack gases, can also be partly eliminated by using some of it to heat the feed water. As the highest steam temperature ordinarily available in the exhaust system is about 212° F., and as the products of combustion leaving the boilers generally have temperatures in the neighborhood of 600° to 700° F., it is evident that on a basis of temperature the hot gases have a decided advantage as a heating medium. On the other hand, the specific heat of the hot gases is low, while exhaust steam can give up all of its latent heat with no change in temperature, so that on a basis of heat available for transmission to the water, the steam has the advantage.

The waste heat in the flue gases is used for feed-water heating in devices known as *economizers*, preheaters, flue gas beaters, etc. These devices are now built in two radically different forms. In one form the heater is mechanically separate from the boiler and is generally located in the flue beyond the boiler damper. In the other the heater is constructed as part of the boiler but is so arranged that water fed into it can be heated by escaping gases before mixing with the water in the main circulating system of the boiler. The separate form of heater is commonly known as an economizer while the one which is essentially part of the boiler is called a preheater, a preheater section, an integral economizer, etc.

For many years economizers were all of the separate type and were made entirely of cast iron: They consisted of cast-iron tubes fastened in groups into cast-iron headers and so arranged in the flues that the tubes stood vertical. The flue gases passed over the external surfaces of the tubes and the water on its way to the boiler flowed through tubes and headers. The apparatus was generally arranged so that water entered at the end at which the gases left and left at the end at which the gases entered. This maintains the greatest available temperature difference between gas and water throughout the entire economizer and is known as a counterflow arrangement.

Cast iron was used both because it is a cheap material and because it is highly resistant to corrosion. Practically all real fuels contain sulphur and some of this sulphur appears in the flue gases as sulphur dioxide. This gas dissolved in water forms sulphurous acid and when further oxidized yields sulphuric acid. If under any conditions the flue gases are cooled to the dew point while passing through the economizer a certain amount of acid or acidulated water is deposited on the external surfaces of the tubes and headers and even with cast-iron corrosion is fairly rapid if such deposits occur in large quantities.

On the other hand, boiler-feed water often contains gases and other impurities which corrode steel rapidly if the water is heated in contact with it. Cast-iron tubes have been found to resist such corrosion to a much greater extent than any form of steel.

The cast-iron economizer is therefore much safer against both internal and external corrosion than is an economizer built of steel.

Economizers are generally connected into the system between the boiler-feed pump and the boiler so that the water within them is at a pressure at least slightly greater than that within the boiler. When boiler pressures were low little thought was given to this fact but as boiler pres-

sures were increased many engineers questioned the practice of using cast-iron economizers under full boiler pressure. Opinions as to the limiting permissible pressure within cast-iron economizers varies, being placed by different engineers at all values between 150 and 250 pounds. Boiler pressures have now reached 300 pounds, and higher in the larger and more modern plants.

Three solutions have been developed. These are:

1. The use of cast-iron economizers at a pressure lower than that in the boiler, a pump drawing hot water from the economizer and pumping it into the boiler. This involves the use of two sets of pumps, one set to force the water into the economizer at a pressure sufficiently high to prevent vaporization when the temperature of the water is raised and the other to raise the pressure from that in the economizer to the pressure of the boiler.

2. The use of steel economizers at full pressure. The use of such equipment involves operation of such character as to guard against the existence of conditions leading to internal and external corrosion.

3. The use of cast-iron and steel economizers in series, the cast-iron economizer being operated under moderate pressure and at the lower temperatures at which the tendency to corrode is generally greatest. This arrangement entails the use of pumps between cast-iron and steel economizers, the latter being operated under full boiler pressure or higher.

At the present time the tendency seems to be toward the use of cast-iron economizers under full boiler pressure in the plants using moderate steam pressures and in the smaller plants while more of the high pressure plants and the larger stations are adopting steel economizers operating under full pressure. The mixed systems and the double pump systems are generally regarded as undesirably costly and complicated.

It is generally conceded that external corrosion of steel economizers can be prevented by never permitting the metal

to have too low a temperature when in contact with flue gases. The limiting temperature varies with the quantity of sulphur in the fuel but exact values are not yet available. A metal temperature in excess of about 125° F. seems to be safe for bituminous fuel with a sulphur content not in excess of 2 per cent while a temperature of 150 to 160° F. seems to be necessary with other fuels having a sulphur content of the order of 5 per cent.

It is also generally conceded that under most conditions internal corrosion can be prevented by satisfactory degasification of the water before entrance to the economizer. All water dissolves air to certain definite quantities determined by temperature and pressure relations if the opportunity is offered and the dissolved oxygen seems to be an active corroding agent when aerated water is heated in contact with steel. The laws governing such solution of air in water are such that the solubility becomes zero when the water is at the point of vaporization. Thus if water under atmospheric pressure is heated to 212° F. under such conditions that the dissolved air can escape and pass off with the vapor generated, complete degasification can be effected. Or if water under any lower pressure is brought to the temperature of vaporization corresponding to that pressure under similar conditions, complete degasification can be brought about. This phenomenon at atmospheric pressure can be observed by raising a vessel of water to the boiling point. The bubbles which form first are bubbles of dissolved gas and it will be noticed that many of them escape from solution before any appreciable quantity of visible vapor, i.e., steam, is formed.

Those plants which are using steel economizers successfully are making provision for adequate degassing and for preventing the degassed water from again dissolving gases before entry to the economizer.

In designing and operating any plant which is to use economizers it is necessary to strike some sort of balance

between the amount of feed-water heating which is to be done by the exhaust steam and the amount which is to be done by gases. With the cast-iron economizers it has been quite common practice to heat the feed with exhaust steam only to that temperature required to guard against external corrosion with steel economizers both internal and external corrosion must be taken into account. The temperatures which have beeen used successfully vary between about 120 and 210° F. depending upon the character of the fuel, the character of the water, the kind of economizer, the arrangement of the plant and other considerations.

Economizers heat the feed water to temperatures varying from about 200° to 350° F. depending upon initial temperature, the extent of the economizers surface, initial gas temperature, and other variables. The temperature of the flue gases drops as the temperature of the water rises, the numerical ratio between the two depending upon relative quantities and specific heats. The temperature of the gases leaving economizers is so low under good operating conditions that induced draft of some sort is generally necessary. For this reason induced draft fans are practically always used when economizers are installed.

If the best results are to be obtained from the use of economizers the heating surfaces must be kept clean. This applies both to the gas side and the water side of those surfaces. In the case of cast-iron economizers it was customary to furnish mechanically operated scrapers for removing the soot deposited on the outside surface of the tubes but many recent installations have omitted such scrapers and substituted provision for cleaning these surfaces with steam lances or soot blowers of some sort. Internal surfaces are cleaned periodically in the same way as the surfaces of boilers, the scale being removed by washing if soft, and by mechanical chipping devices if hard.

The great amount of heat carried away by the flue gases when economizers are used would lead one to assume that

economizers should always be installed. This is, however far from the truth. Economizers are costly and their installation involves the provision of space, and supporting structure, additional flues, amplified draft apparatus, additional piping, etc. All this represents increased investment, bringing certain definite capital charges. Moreover, the use of economizers brings in certain operating and maintenance expenses not incurred if they are not installed. The advisability of using economizers can be determined only after a proper balance of increased charges against the money value of the fuel saving to be expected. In general, the higher the cost of the fuel and the more nearly the plant runs throughout the year at full capacity, the better the chances for a net saving by the installation of economizers.

PROBLEMS

1. Determine the heat lost in the chimney gases per pound of coal in a plant operating under the following conditions, and express the loss as a percentage of the heat value of the coal. The coal has a calorific value of 14,000 B.t.u. per pound; the temperature of the gases leaving the boiler is 570° F.; 20 lbs. of gas result from each pound of coal burned; the mean value of the specific heat of the gases is 0.245; and the temperature of the air entering the furnace is 75° F.

2. Determine the quantity of heat which could be obtained from the gases of Prob. 1 by using an economizer to reduce their temperature to 250° F. What percentage of the heat value of a pound of coal does this saving represent?

3. The boilers of a certain plant produce 100,000 pounds of steam per hour when the plant is operating at full load. The steam-driven auxiliaries consume 10% of this steam. Steam is generated at a pressure of 175 lbs. per square inch gauge, and is superheated 150° F. The main units operate condensing and the condensate leaves the condensers at a temperature of 75° F. The auxiliaries operate non-condensing and exhaust their steam at atmospheric pressure and with a quality of 92%. The coal used has a calorific value of 13,850 B.t.u. The boiler efficiency $\left(\frac{\text{Heat given water and steam}}{\text{Heat in fuel supplied}}\right)$ is 75%.

(*a*) Determine the amount of coal which would have to be burned per hour if the steam exhausted from the auxiliaries were thrown away and make-up water at a temperature of 50° F. were used in its place. The condensate from the condensers of the main unit is assumed to be returned to the boiler after being mixed with the make-up water.

(*b*) Determine the amount of coal which would have to be burned per hour if the auxiliary exhaust were used to heat the condensate from the main units in an open heater and if the operation of the plant were so perfect that no make-up water had to be added.

CHAPTER XIX

BOILER-FEED PUMPS AND OTHER AUXILIARIES

163. Boiler-feed Pumps. The pumps used for forcing the feed water into boilers may be of reciprocating or centrifugal construction and may be driven by reciprocating steam cylinders, by small steam turbines or by electric motors.

Steam-driven pumps are very wasteful, often using over 100 lbs. of steam per horse-power hour. It would therefore seem more economical to use motor-driven pumps in electric-power stations, as the large power units will generate electric power with a consumption of from 10 to 25 lbs. of steam per horse-power hour and the motor efficiency will generally be over 80 per cent. There is, however, another point which must be considered. The exhaust steam from small engines operating boiler-feed pumps can be used for heating the feed water as described in the last chapter, and thus the poor economy of these units is of little significance; practically all heat exhausted can be returned to the boiler in the boiler feed if desirable. As a result of this consideration, coupled with others of less importance, nearly all boiler-feed pumps and other similar auxiliaries are steam driven unless there are so many that there would be more exhaust steam than could be absorbed by the feed water.

There is at present a marked tendency toward the use of turbine-driven, centrifugal pumps for boiler feeding, in place of those driven by reciprocating steam units. The turbine type has several advantages, the more important being:

(1) No oil in exhaust steam, so that latter is well adapted to use in all forms of feed-water heaters;

(2) Higher speed because of continuous flow of water and continuous rotation of mechanical parts, thus making possible great decrease in size for a given amount of work, and

(3) Better pump characteristics for this sort of work.

The Duplex Steam Pump. The great majority of reciprocating steam pumps used for boiler-feed purposes are of the duplex pattern, one design of which is shown in Figs.

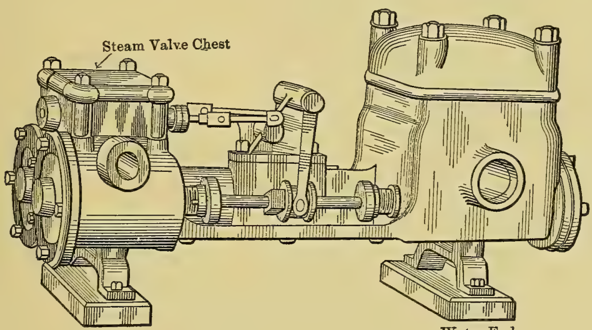


FIG. 240.—Duplex Steam Pump.

240 and 241. Two steam cylinders are arranged side by side, their piston rods extending into similarly arranged water cylinders and carrying water plungers or pistons as shown in Fig. 241. As there is no rotating shaft in a pump of this kind, the steam valves cannot be operated by eccentrics as is common with steam engines. For the purpose of operating these valves, bell cranks, pivoted near the center of length of the pump, are provided. These are arranged so that the long arm of one bell crank engages a collar on the piston rod of one steam cylinder and the short arm operates the valve gear of the other steam cylinder. The motion of the valve of one cylinder is therefore derived

from the piston motion of the other cylinder. The steam pistons are practically 180° out of phase, one moving out while the other moves in.

Practically no expansion of the steam is obtained in the cylinders of pumps of this type. They operate on the rectangular cycle described in an earlier chapter and are correspondingly wasteful in their use of steam.

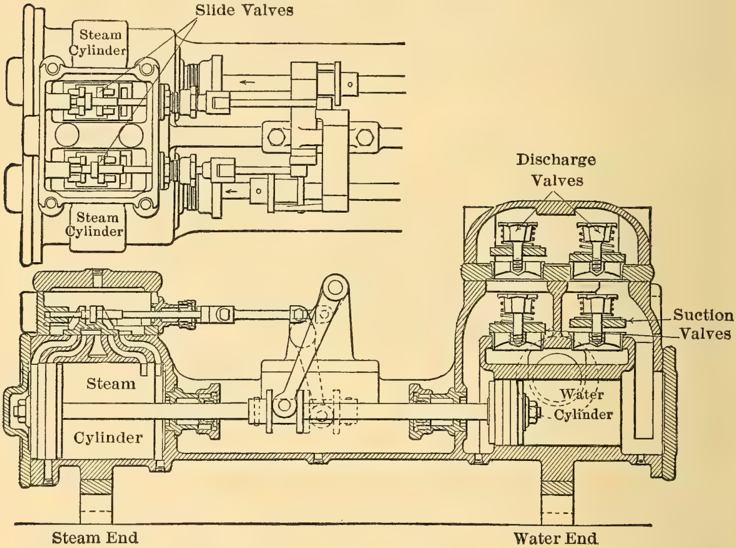
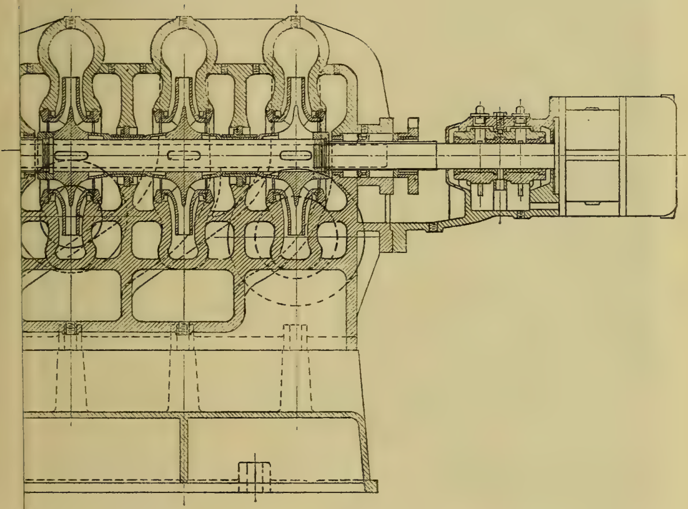


Fig. 241.—Duplex Steam Pump.

A turbine-driven, centrifugal boiler-feed pump is shown in section in Fig. 242. The turbine is a multistage arrangement of the impulse type, having one Curtiss wheel at the high pressure end. The pump is a three-stage device, the first stage discharging to the suction of the second stage and the second stage discharging to the suction of the third stage. By multistaging in this way any desired boiler-feed pressure can be obtained with moderate rotative speed and diameter.



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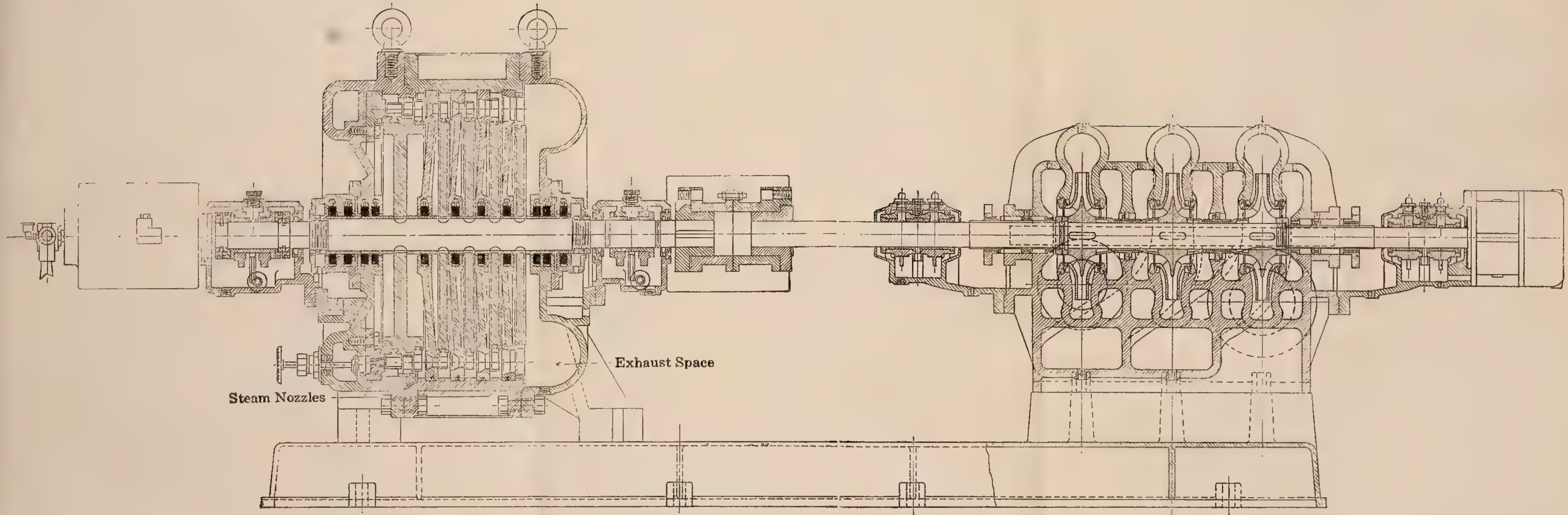

FIG. 242.—Steam Turbine-driven Centrifugal Boiler Feed Pump.

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164. The Steam Injector. On steam locomotives and in other portable steam plants, as well as in many small

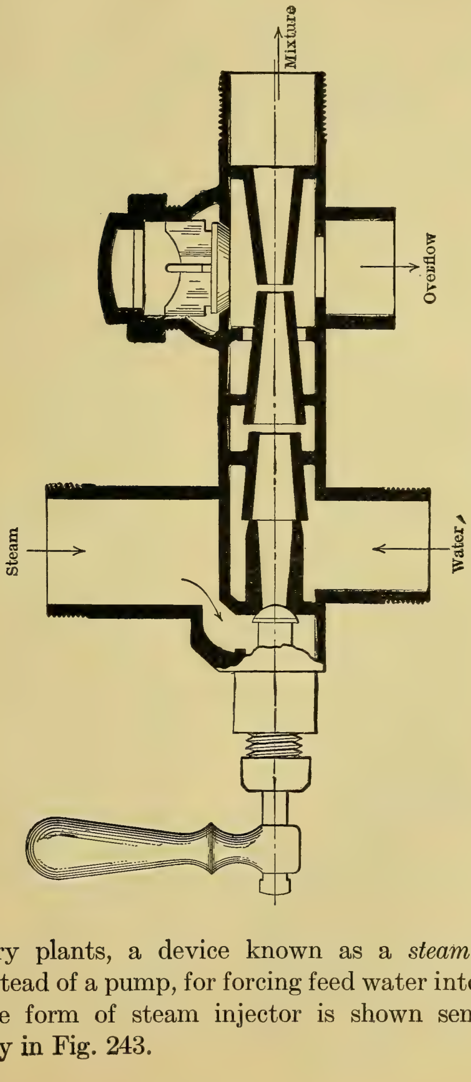


FIG. 243.—The Steam Injector.

stationary plants, a device known as a *steam injector* is used, instead of a pump, for forcing feed water into the boiler. A simple form of steam injector is shown semi-diagrammatically in Fig. 243.

Steam from the boiler flows through the steam nozzle and expands from boiler pressure to a very low pressure, thus acquiring a high velocity at the expense of the heat energy which it brings from the boiler. At the end of the nozzle it mixes with water and imparts to that water some of its kinetic energy, so that the mixture moves into the small end of the delivery tube with a high velocity. By the time it has reached that point, practically all the steam has been condensed, and, as the sectional area of the delivery tube increases, the velocity of the liquid decreases with a corresponding increase in pressure according to Bernoulli's theorem. In properly designed apparatus, the resultant pressure is great enough to force the mixture of water and condensed steam into the boiler against boiler pressure.

The space at the end of the steam nozzle is maintained at a low temperature by the feed water flowing through it and the pressure of the steam is therefore very low at this point, being less than atmospheric in most cases. Atmospheric pressure is therefore able to force water up the suction pipe if the "lift" is not too great, and when once started such a device can therefore "raise" its own water as well as deliver it against pressure.

It is interesting to note that the efficiency of this apparatus is almost 100 per cent on a heat basis. All heat not radiated from the apparatus is returned to the boiler in the mixture of condensed steam and feed water and, as the external surface is very small, very little heat is lost by radiation.

165. Separators. Two kinds of separators are used in steam plants: (*a*) the *oil separators* already referred to for separating oil from exhaust steam, and (*b*) *steam separators*, which separate water from steam.

As it is impossible entirely to prevent radiation from steam pipes, it follows that condensation will occur in any pipe line which carries saturated steam. Water is also formed in the cylinders of reciprocating engines not supplied

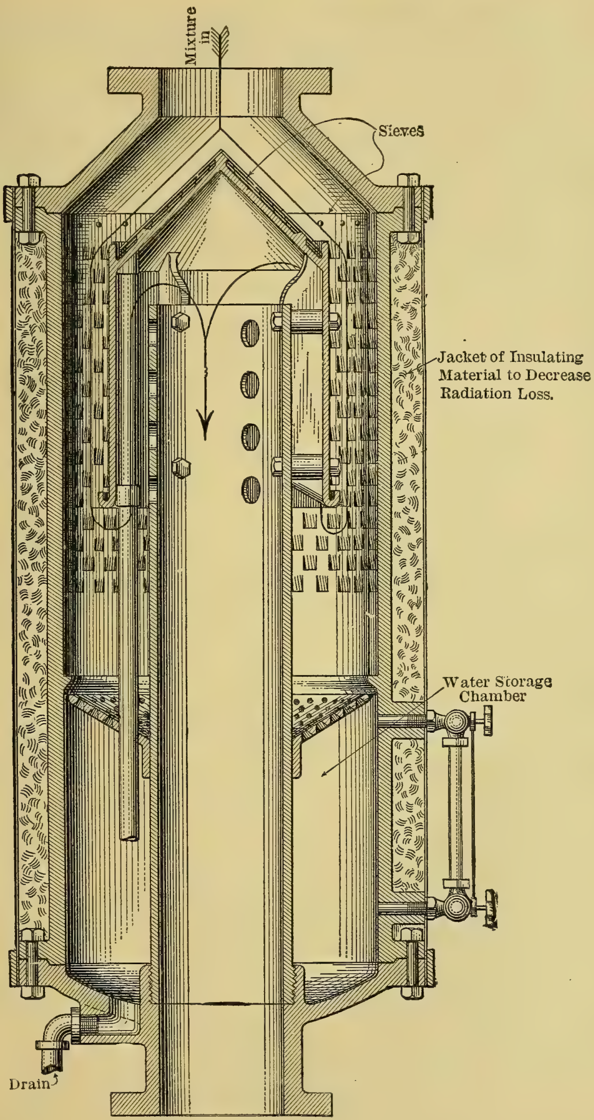


FIG. 244.—Steam Separator

with very highly superheated steam, and much of it is generally present in the exhaust of the high and intermediate cylinders of multiple-expansion engines.

A small amount of water can be passed through the cylinder of a reciprocating engine without mechanical damage, but it probably causes a loss of heat by clinging to the walls and assisting in the heat interchanges which always occur. Large quantities of water are apt to cause mechanical damage, as water is inelastic, and if more of it is trapped in a cylinder end than can be contained in the clearance, something must give way when the piston reaches the end of its stroke.

It is customary to separate as much as possible of the water of condensation before admitting steam to the cylinder. The separators used are built in many different shapes and types, but practically all depend upon two principles. These are:

(1) Water is much more dense than steam, and if a stream of a mixture of water and steam be made to travel in a curve, the water will therefore collect at the outside of the curve, and

(2) Water brought into violent contact with metallic surfaces "wets" them and has a tendency to adhere thereto.

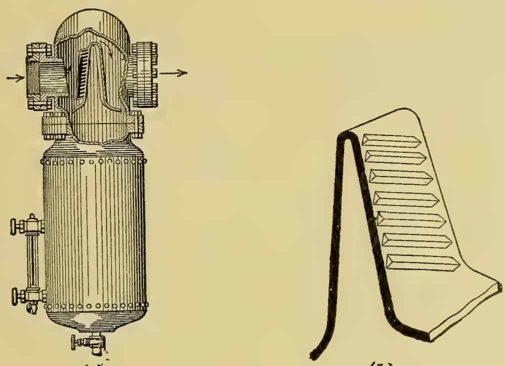
In steam separators the stream of mixture is therefore made to change its direction of flow suddenly and to impinge upon baffles in such a way that the greater part of the liquid is caught and drained off.

One form of separator is shown in Fig. 244. The mixture impinges on seives in the first part of its passage through the separator, part of the water passing through the openings and draining to the reservoir at the bottom of the device. Ridges and troughs catch all water separated and guide it to drains leading to the reservoir so that no water which is once deposited is again picked up by steam.

Another form of separator is illustrated in Fig. 245. The steam impinges upon the inverted V-shaped casting

and water caught on the projecting ridges drains toward the sides and then downward into the receiver, while the steam passes on as shown.

166. Steam Traps. In the separators just described, there is a constant accumulation of water which must be drained off periodically if the entire device is not to fill up and become inoperative. Similarly there is a constant accumulation of liquid in steam jackets, in receivers of multi-expansion engines and in low points in steam lines.



(a) (b)

FIG. 245.—A Steam Separator.

167. Steam Piping. There is a great deal of piping of various kinds in all steam plants and the financial success or failure of a plant often depends upon this apparently insignificant item. It is beyond the limits of a book of this scope to consider the many different forms of piping and the many different ways in which apparatus may be connected. This is a study in itself and one of great importance.

It should be noted, however, that all of the following points must be kept in view when designing and installing piping and that that installation which most nearly meets all these requirements may be regarded as the best.

(1) The various lines should conduct the materials flow-

ing through them with the minimum loss of pressure and with the minimum loss (or gain) of heat.

(2) The pipe lines should be so constructed as to make failure of a dangerous sort, from expansion and contraction, water hammer and such, most unlikely if not impossible.

(3) All connections should be so made that the careless manipulation of valves cannot cause an accident.

(4) The number of flange and screw connections and the number of valves and fittings should be reduced to the minimum, as they are often sources of weakness and are always costly.

(5) The entire layout should be so arranged that interruption of service because of pipe, or valve, failure is (as nearly as possible) impossible.

(6) The cost of the system should be as small as it can be made, consistent with the other requirements.

It is almost unnecessary to say that all of these desirable ends are never attained in any plant. A compromise must always be made in order to bring the cost within reasonable limits, but most of the recent installations show a tendency toward better design in this part of the plant and a consideration of reliability and safety far in excess of what was formerly customary.

CHAPTER XX

PERFORMANCE OF STEAM POWER EQUIPMENT

168. Meaning of Performance. The term performance is used in a general sense in engineering and refers to the extent to which, or the way in which, a material or piece of apparatus or a structure performs, that is, the extent to which it meets expectations or the way in which it compares with other similar things. The performance of steam power equipment of different sorts is measured in different ways, depending on the purpose in view.

Thus the performance of a given engine with respect to a theoretical engine might be measured by comparing the quantity of steam used by the real engine when doing a certain amount of useful work, with the quantity of steam which would be required if the theoretical incomplete-expansion cycle could really be attained and utilized. Or in other cases the performance might merely be expressed in pounds of steam used per delivered or brake horse-power hour or per indicated horse-power for comparison with other engines of like or unlike characteristics.

Or, the efficiency of a boiler might be determined by dividing the heat put into water and steam (i.e., useful output or result) by the heat in the fuel fired (i.e., input). The numerical value thus obtained would be a measure of the performance of the boiler with respect to the extent to which it utilized thermal energy.

Or, the engineer might be interested in a complete plant generating electrical energy and he might desire to compare it with other plants doing the same thing and particularly to compare it with respect to its thermal efficiency. He might determine the overall efficiency by dividing the

thermal equivalent of the electrical output by the heat value of the fuel supplied to produce that output, or he might determine the number of thermal units supplied in fuel for each kilowatt-hour produced. Either expression would serve as a measure of the performance of the plant in this respect.

Or an engineer might want to compare the performance of one boiler room with another with respect to the total of all costs. For such purposes he might use the total cost (including fuel, water, supplies, labor and all maintenance charges) per thousand pounds of steam produced. Such a figure would serve as a measure of performance for such a purpose.

It is impossible to consider in a book of this kind all or even many of the different measures of performance used in connection with steam power equipment. Attention will be limited to some of the most common which all fall under the heading of efficiencies.

Reduced to its simplest terms the function of a steam power plant consisting of steam boilers, prime movers and all associated apparatus, equipment and buildings is to produce a certain amount of useful mechanical energy as a result of expending a greater quantity of heat which is liberated by burning fuel. The extent to which it succeeds in a thermal sense is measured by the overall thermal efficiency of the plant, that is the quotient obtained by dividing the heat equivalent of the useful mechanical energy by the heat supplied in fuel. This thermal efficiency is not, by itself, a complete measure of the performance, since high efficiency does not necessarily mean lowest cost of power. This fact should never be lost sight of.

The overall thermal efficiency of the plant is a composite made up of a number of other thermal efficiencies, principally those of the boilers and those of the prime movers. High boiler efficiency combined with high prime mover efficiency does not necessarily mean high overall

efficiency of the plant because the overall efficiency of the plant is also influenced by the ways in which auxiliary equipment is arranged and operated. However, with other things equal, or nearly so, the thermal efficiencies of boilers and prime movers will determine the over-all performance.

These facts lead to frequent determinations of such thermal efficiencies for the purpose of guiding manufacturers of such equipment, designers and builders of power plants and operators of power plants. It is the purpose of this chapter to consider briefly how such thermal efficiencies are determined. In order to be logical the boiler will be considered first and the prime mover second.

169. Determination of Boiler Performance. The thermal performance of the boiler is most commonly expressed as overall thermal efficiency. This means the ratio of heat put into water and steam in the boiler to the heat in the fuel supplied to the boiler furnace. Quite obviously some means must be available for determining not only the total quantity of fuel supplied in a given time but also the heat value, or calorific value, of that fuel. The quantity of fuel supplied is determined by weighing the fuel if it is solid and by means of appropriate meters if it be liquid or gaseous. Weighing is also sometimes used with liquid fuels.

The calorific value of the fuel is determined by testing carefully taken samples in fuel calorimeters which will be described in a later paragraph.

With the heat supplied determined in this way it is still necessary to measure the useful output. This is determined from (*a*) the quantity of steam leaving the boiler and (*b*) the heat added to it in the boiler. The quantity of steam produced is usually determined by measuring the water supplied the boiler, either by weighing or by means of carefully calibrated meters and then carefully guarding against loss through blowoff connections or other openings into the boiler.

The temperature of the feed water is measured by means

of an accurate thermometer as it enters the boiler to obtain a measure of the heat supplied in the feed water. The pressure and temperature or quality of the steam leaving are also determined. If the boiler is producing superheated steam, pressure and temperature give all necessary data for determining the heat above 32° F. If the boiler is producing saturated steam, pressure and quality give the necessary information.

Temperature of superheated steam is determined by means of a thermometer immersed in liquid contained in a well projecting into the interior of the steam pipe. Quality of saturated steam is determined by means of steam calorimeters which are described in a later paragraph.

170. Fuel Calorimeters. The calorific value of fuels is determined by means of instruments known as fuel calorimeters, which were briefly referred to in Chapter XVI. The fuel calorimeter is a device in which a known quantity of fuel can be completely burned under such conditions that all the heat liberated can be measured.

The bomb calorimeter is the standard instrument for determining the calorific value of solid fuels such as coal and it is also sometimes used with liquid fuels. A vertical section through one make of bomb calorimeter is shown in Fig. 246.

In this figure the letter *a* designates the bomb. This is a heavy walled vessel, generally made of steel and lined with a non-corrosive material such as nickel. The non-corrosive lining is used to prevent attack by acids formed during combustion of the fuel.

Within the bomb is supported a small shallow vessel known as the crucible. This crucible receives the sample of fuel which is to be burned in the calorimeter. The sample is finely ground and sometimes dried before insertion in the crucible and an amount weighing about one gram is ordinarily used.

When making a determination the sample is placed

in the crucible, the crucible is placed on its support in the lower half of the bomb, the upper half of the bomb is put in place on the lower half and the two halves are locked together so as to form a gas-tight joint. The necessary oxygen for supporting combustion is insured by filling the bomb with oxygen under a pressure of several hundred pounds.

Combustion is started electrically and it is then self continuing until all of the combustible is consumed. Two different methods are used for initiating combustion. In one, a fine iron wire partly immersed in the coal in the crucible is heated by passing current through it until it ignites and burns and thus ignites the coal. In the other a platinum wire similarly located is heated in the same way. It ignites the coal but is not itself consumed.

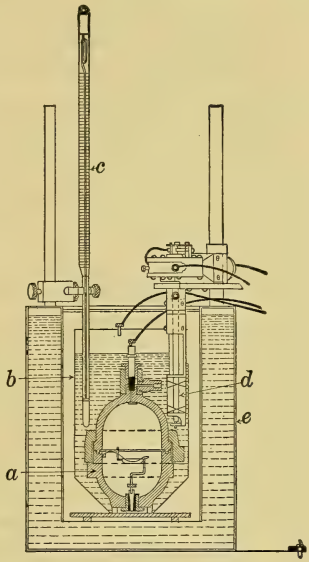


FIG. 246.—A Bomb Calorimeter.

When the bomb has been prepared for a determination by the insertion of the sample and the necessary oxygen it is placed in the vessel *b*, figure 246, which is then filled with a known weight of water. A thermometer, designated by *c*, in the figure is provided for reading the temperature of the water and a motor-driven stirrer designated by *d*, is arranged to circulate the water and thus maintain it at a uniform temperature.

The bomb, the water around it, the vessel *b*, the thermometer and the stirrer really form the calorimeter and could be used as such. However, it is customary to supply another vessel with double bottom and double walls to serve

as a thermal shield for the calorimeter proper. In the type shown in Fig. 246 this double-walled vessel is designated by *e* and is filled with water.

It is obvious that if a sample of fuel is burned in a device such as that shown, the heat which is liberated will heat the contents of the bomb, the bomb itself, the water in which it is immersed, the container or vessel *b* and parts of the thermometer and the stirrer. If no heat were used in any other way it would be possible to determine the total quantity liberated during combustion from the respective weights of the materials heated and the temperature rise as determined by the thermometer *c*.

In fact, however, the interchange of heat between the calorimeter and its surroundings and certain other phenomena make it necessary to add complications to what would otherwise be a simple procedure.

The interchange of heat between the calorimeter and its surroundings is spoken of as "radiation" and a "radiation correction" is determined during each test. The determination is made by observing the rate of change of temperature of the water as indicated by the thermometer *c* after assembling the apparatus and while it is attempting to reach the same temperature as that of its surroundings. When the rate of change has been established by several readings at regular intervals the charge is ignited. Temperature readings are continued at certain definite intervals while the temperature of the water rises to a maximum due to the heat liberated within the bomb and then begins to drop again at a regular rate due to loss of heat to the surroundings.

A radiation correction is calculated from the respective rates of change of the temperature before combustion and after the attainment of maximum temperature. By means of this correction, the temperature rise resulting from combustion (as read on the thermometer) is corrected to what it would have been had it been possible to thermally

isolate the calorimeter. With this corrected temperature rise it is then possible to calculate the total quantity of heat liberated within the bomb.

Instead of using the respective weights of all of the different materials of the calorimeter in this calculation it is customary to determine, once for all, what is known as the "water equivalent" of the calorimeter. The water equivalent is that weight of water which would experience the same increase in temperature as does the real calorimeter with any given supply of heat. This water equivalent in pounds, multiplied by the temperature rise in Fahrenheit degrees (after correction for radiation) would then give directly the British thermal units liberated within the bomb.

There are several different methods of determining the water equivalent. That which is generally considered the most satisfactory is based upon the combustion within the bomb of a known weight of material of known calorific value. When this is done, and the necessary radiation correction is made, part of the known amount of heat liberated will be accounted for by the observed temperature rise of the known weight of water in the calorimeter. The remainder is accounted for by the same temperature rise of the "water equivalent" of the rest of the calorimeter. The water equivalent can therefore be calculated with the data available.

In preceding paragraphs reference has been made to the "heat liberated within the bomb." This wording was used because the heat liberated within the bomb is not necessarily the same as the heat resulting from combustion of the fuel within the bomb. For example, if an iron wire is used for igniting the fuel and if this iron wire burns as it generally does, its combustion supplies heat in addition to that supplied by the fuel.

In any case, such additional heat supplies must be evaluated and deducted from the total absorbed by the

calorimeter in order to obtain the heat liberated by the combustion of the fuel.

The bomb calorimeter, when properly constructed and used, is the most accurate instrument available for determining the calorific value of solid fuels. It is, however, an instrument which requires very careful manipulation and it does not lend itself to rapid determinations. For such reasons many other types have been produced and some of them are capable of giving results which are sufficiently accurate for many commercial purposes.

An entirely different type of calorimeter known as the Junker calorimeter is most commonly used for determining the calorific values of gaseous fuels. It can also be used with many liquid fuels. Such a calorimeter arranged for use with gaseous fuel is shown in Fig. 247. The fuel gas enters the system through a very accurate meter indicated by *a* in the figure. From the meter it passes to the gasometer *b* which serves to maintain constant gas pressure. From the gasometer it passes through the tube *c* to a burner which projects into the center of the large cylinder *d* which is the calorimeter proper.

The gas is burned within the cylinder *d* by means of air admitted through adjustable openings in the bottom of the cylinder. The products of combustion first pass upward to the top of the cylinder. They then flow downward in tubes of small diameter which pass through a space between the cylinder *d* and an inner cylindrical wall parallel to it. They then pass out of the instrument through the nozzle *e* which contains an adjustable damper for controlling the rate of flow.

Water enters the instrument through the tube *f* being delivered to a small box or head tank *g*. An overflow pipe *h* serves to maintain a constant level in this head tank. From the head tank the water flows down the pipe *i* and into the calorimeter vessel through the regulating cock *j*. Within the calorimeter vessel it flows upward in the space

between the two concentric cylinders previously mentioned, entirely surrounding the small diameter pipes which carry the products of combustion downward. The water finally leaves through the discharge tube *k* which spills it into a graduated measuring cylinder *l* when a run is being made.

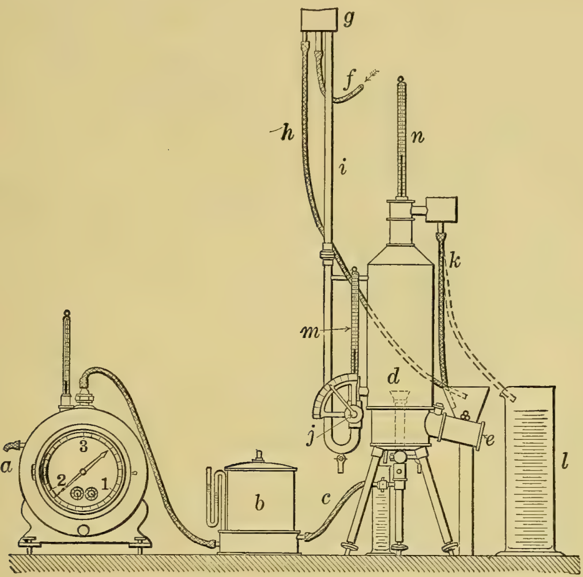


FIG. 247.—Junker Calorimeter.

A thermometer *m* gives the temperature of the water entering the calorimeter and another one *n* gives the temperature of the water leaving.

With the quantity of water and its entering and leaving temperatures determined, the quantity of heat liberated is calculated easily. The quantity of fuel used is measured by the meter *a* and it is therefore possible to calculate the heat value per unit of volume or weight.

The instrument is commonly adjusted so that the products of combustion leave the nozzle *e* at the same temper-

ature as that with which the gas and air enter the instrument. Under such conditions the heat value determined is very nearly the higher heat value and it is commonly so regarded.

171. Steam Calorimeters. The devices known as steam calorimeters are used for determining the quality of saturated steam and, occasionally, the degrees superheat of superheated steam. They do not measure heat as do the fuel calorimeters. Since they are not heat measures the term calorimeter is really a misnomer. They are really quality meters.

There are many different varieties of steam calorimeters, some very crude and some very exact instruments. Only the three most common types will be considered here. These are the *Separating Calorimeter*, the *Throttling Calorimeter* and the *Universal Calorimeter*.

The *Separating Calorimeter* is, as its name implies, a device in which quality is determined by separating out the water and determining its proportion of the total. A section through one form of separating calorimeter is shown in Fig. 248.

Steam enters the device at the top passes through a small separating device indicated by *a*. The water which is carried in suspension separates out at this point and gravitates to the bottom of the inner vessel or cylinder indicated by *b* in the figure. The steam which has given up its suspended moisture passes through the small openings indicated, flows through the jacket space *c* and out at the bottom of the calorimeter through a tube *d* which leads it beneath the surface of water contained in the vessel *e*. The opening through which the steam leaves the jacket of the calorimeter is made of small diameter to maintain the pressure in the jacket at practically the same as that within the calorimeter. The jacket temperature is thus maintained.

A water glass or gauge glass, *f*, is connected into the water collecting space of the calorimeter and serves to indi-

cate the height of the accumulating water. A slide, *g*, which can be moved up and down on this glass indicates the weight of water within the calorimeter by reference to a scale, *h*.

The instrument is used by first allowing a sample of the steam under test to flow through it continuously and

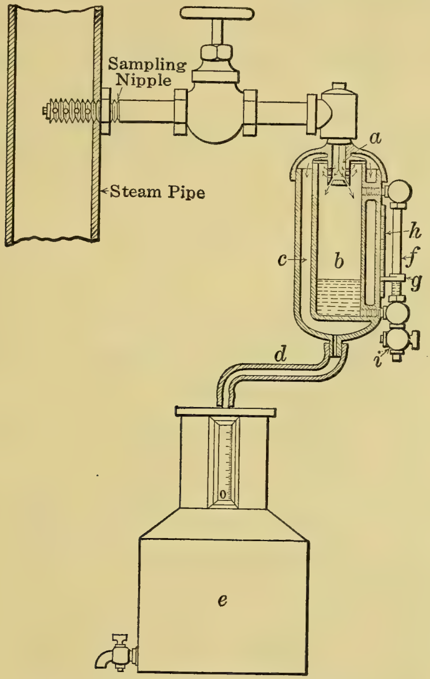


FIG. 248.—Separating Calorimeter.

to run to waste until the entire instrument has been raised to working temperature. Water which collects in *b* during this period is drained down to a convenient point by bleeding through the cock *i*. The weight of water in the container *e* is determined by weighing or the height is noted by reference to the scale on the neck of the vessel. The height of water in the glass *f* is noted and then the dis-

charge tube d is dropped as quickly as possible into the container e.

The instrument then continues in operation until enough steam has passed through to give reasonable accuracy to the determination. The flow of steam through the instrument is then discontinued, the tube e is withdrawn from the container and the quantities of water separated in the calorimeter and of steam condensed in the container e are determined.

If the effect of radiation and other possible sources of error is neglected the quality of the sample is then

$$x=\frac{W}{W+w}, \qquad (103)$$

in which

x = quality of sample expressed as a decimal fraction;
W = dry steam leaving calorimeter and condensed in container, measured in pounds;
w = water collected in calorimeter, measured in pounds.

It is customary to minimize heat loss from the instrument and from the pipe bringing the sample to the calorimeter by covering all parts with hair felt which is an excellent thermal insulator. For exact work it is, however, customary to determine the heat lost by "radiation" and to correct for this amount. This is conveniently done by expressing this heat loss in terms of the amount of water which will collect in the calorimeter in a given time and then using the formula for quality in the following form:

$$x=\frac{W+R}{W+w}, \qquad (104)$$

in which

x, W and w have the same significance as before, and
R = weight in pounds of that part of total liquid

separated in calorimeter which must have condensed to supply the heat lost from the instrument and connections.

An improved form of separating calorimeter is shown in section in Fig. 249. The improvement consists in the use of a calibrated nozzle at the bottom of the jacket and a gauge graduated to read flow of material in a given time. When this instrument is properly calibrated and used it is capable of giving accurate results but the calibration should be checked at intervals by condensing the discharged steam.

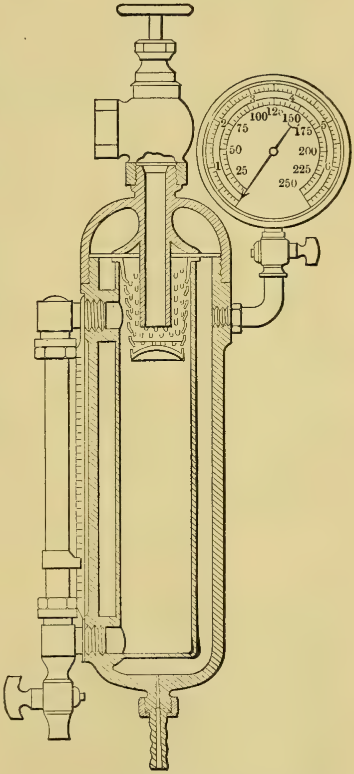


Fig. 249.—Separating Calorimeter.

The separating calorimeter can be used satisfactorily to determine the moisture in steam over the entire commercial range of quality. However, with very small moisture contents it is necessary to work very carefully and to apply several corrections to obtain accurate results. For this reason many engineers prefer to use the throttling calorimeter under such conditions.

A *throttling calorimeter* is shown in section in Fig. 250. Steam enters through a small nozzle a and leaves through a much larger opening b from which it is carried to any

convenient point for disposal. The nozzle at *a* has so small an opening that steam cannot flow into the interior of the calorimeter fast enough to build up any great pressure. Under ordinary conditions of use the pressure within the calorimeter is equal to atmospheric or a fraction of an

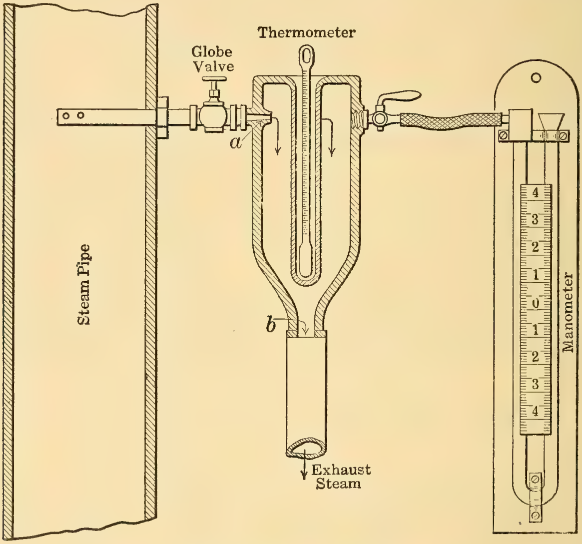


FIG. 250.—Throttling Calorimeter.

inch of mercury above atmospheric. The steam flowing through the nozzle *a* therefore expands from the pressure in the pipe leading to the calorimeter to the much lower pressure within the calorimeter, a process analogous to that occurring in the nozzles of steam turbines as described in Chapter XIII. There is an important difference between the two cases which will appear in later paragraphs.

If the calorimeter is assumed to have been in use for some time, so that all metal parts are heated up to working temperature and if loss of heat to surroundings be assumed to be zero, all of the energy in the steam entering through

the nozzle must leave with the steam issuing from the large opening at the bottom of the calorimeter for the simple reason that there is no other way in which it can escape.

This is the principle on which the calorimeter works and it can be put in the form of an equation thus, energy entering = energy leaving. The energy in the entering steam consists of two parts, that determined by its pressure and quality and determinable from the steam tables and that due to the velocity with which it flows in the small pipe leading to the calorimeter. This velocity is so small that it can be neglected without sensible error and it is always so neglected. The energy entering the calorimeter is therefore taken as that determined by the state of the steam and is equal to $q+xr$ per pound of wet saturated steam as given in equation (15) page 34.

The energy in the steam leaving the calorimeter is similarly that due to the state of the steam and that due to its velocity of efflux from the calorimeter. This velocity is always small under normal conditions of use so that it can be neglected. The energy determined by the state of the steam is therefore taken as that in the steam leaving and as that which must be equal to the heat above 32° F. in the entering steam. It follows from these facts that the fundamental equation of the throttling calorimeter becomes

Heat above 32° F. in entering steam = heat above 32° F. in leaving steam if heat loss by " radiation " is neglected.

If the steam tables are consulted it will be discovered that for saturated steam the total heat above 32° F. at a pressure near atmospheric is always lower than that for a higher pressure. In fact, if the higher pressure is 50 pounds or more as is common in power plant work the difference will be comparatively great. It is therefore true that if there is not too much water in the high-pressure steam flowing to the calorimeter the low-pressure steam leaving it will have to be superheated to contain the necessary

heat above 32° F. This is the only condition under which this form of calorimeter can be used. Under such conditions, the fundamental equation of the calorimeter becomes

$$[xr+q]_1=[y+C_{pm}(t_s-t_v)]_2, \quad . \quad . \quad . \quad . \quad . \quad (105)$$

in which subscript 1 denotes values for entering steam, subscript 2 denotes values for leaving steam, t_s is temperature of the superheated steam in the calorimeter and t_v is the temperature of vaporization corresponding to the pressure in the calorimeter. From this equation, the quality x must be

$$x=\frac{[\lambda+C_{pm}(t_s-t_v)]_2-q_1}{r_1}. \quad . \quad . \quad . \quad (106)$$

The calorimeter is used by allowing steam to blow through it and noting the pressure of the sample outside the calorimeter, the pressure inside the calorimeter, the barometric pressure and the temperature of the steam inside the calorimeter as indicated by the thermometer immersed in liquid in the thermometer well projecting into the interior of the calorimeter. The pressure within the calorimeter is expressed in pounds absolute and the corresponding temperature of vaporization is determined from the steam tables. If this is less than the temperature indicated by the thermometer the steam in the calorimeter is superheated and the quality of the entering steam may be determined. It is only necessary to convert the pressure of entering steam into pounds absolute, take the necessary values from the steam tables and substitute in equation (106) above.

If the steam entering the calorimeter is superheated instead of wet there will be no indication of such a fact until equation (106) is solved. The value obtained for x will then be greater than 1. When this happens it is possible to determine the degree of superheat, if desired. For this purpose take from the steam table the total heat of

dry saturated steam for the pressure at which the steam enters the condenser and subtract it from the total heat of the steam leaving the condenser, i.e., the right-hand member of eq. (105). The difference is the amount of heat contained in the entering steam by virtue of its superheat and must be equal to $[C_{pm}(t_s-t_v)]_1$. The value of t_v is given in the steam tables but C_{pm} and t_s are unknown. A solution must therefore be found by trial and error. That is, some value of t_s is assumed, the corresponding value of C_{pm} is obtained from Fig. 14 and both are substituted in the expression last given and the result compared with the amount of heat to be accounted for. If the difference is greater than permissible, another initial assumption is made, and this is continued until the desired degree of accuracy is obtained.

All calculations outlined above for determining the quality or initial superheat by means of observations made with a throttling calorimeter can be eliminated by using the Mollier Chart, previously given as Fig. 154. It has been shown that values are determined merely by assuming the heat above 32° F. to be the same for entering and leaving steam and then solving for quality of entering steam after obtaining all other values by observation and from the steam tables.

The Mollier Chart is so drawn that any change occurring with constant heat above 32° F. is represented by a vertical line. If the condition of the low pressure steam in the calorimmeter (and leaving the calorimeter) is spotted on this diagram it will be found to be somewhere near the middle of height (about 15 pounds) and to the left of the saturation curve. A line dropped vertically from that point to the curve for the initial steam pressure will show at its intersection with that curve the initial condition of the steam.

It is obvious that if the relative locations are such that this vertical line crosses the saturation curve the initial condition must have been that of wet steam, whereas if

it does not cross, the initial condition must have been that of superheated steam.

The difference in the behavior of steam flowing through a turbine nozzle and that of steam flowing through a throttling calorimeter nozzle can now be brought out. In the chapter on Steam Turbines it was shown that that which happens in a turbine nozzle is pictured by a horizontal line drawn from left to right on the Mollier Chart. Inspection of the chart will show that this means a change which will reduce superheat or increase wetness. And yet in the case of a similar nozzle in a throttling calorimeter we seem to meet a different condition.

That which occurs in the nozzle is substantially the same in both cases. It is in what happens after the steam leaves the nozzle that the difference occurs. In both cases expansion within the nozzle is essentially isentropic and represented by a horizontal line drawn from left to right on the diagram. The steam pressure drops, the heat above 32° F. decreases and the difference appears as kinetic energy of motion. In the case of a turbine nozzle the turbine wheel removes this kinetic or velocity energy from the steam and thus leaves it with the smaller heat above 32° F. with which it flowed out of the nozzle.

In the case of the calorimeter the kinetic energy is converted back into heat as the jet of steam is brought to rest by bombarding its surroundings. The heat above 32° F. thus becomes the same at the end of the whole process as it was at the beginning merely because the kinetic energy developed in the nozzle is not removed as it is in the case of the turbine. It is obvious that a complete representation of what happens in the calorimeter is more complicated than the simple vertical line used above to obtain a solution. However, it is not necessary to study the intermediate steps of the process as initial and end conditions give all that is necessary for the determination of initial quality or superheat.

The throttling calorimeter has certain definite limitations. It will work with any initial superheat but if the initial quality is too low the steam will not be superheated in the calorimeter and the initial quality cannot be determined. This can be shown by means of the Mollier Chart.

Assume, for instance that the intitial steam pressure is 50 lbs. absolute, that the initial quality is 95 per cent and that the pressure within the calorimeter is 15 lbs. absolute. Locate the intersection of the 50 lb. and 95 per cent line and run vertically upward on the chart to intersection with the 15 lb. line. The chart indicates that the steam will still contain moisture when expanded into the calorimeter. In fact, the chart indicates that the initial quality will have to be about 97.5 per cent to just dry the steam in the calorimeter.

It is not safe to use a throttling calorimeter with less than about 10° superheat in the calorimeter so that for 50 lbs. absolute steam pressure, the initial quality would have to be about 98 per cent or higher to make the throttling calorimeter usable.

The percentage of water at which the throttling calorimeter becomes inoperative increases with the initial pressure so that at 200 lbs. absolute a quality as low as 95 per cent can be determined with this instrument.

The *Universal Calorimeter* is a combination of a throttling calorimeter and a separating calorimeter. It is so arranged that the steam passes first through the throttling calorimeter. If it is superheated in this part of the apparatus nothing further is required. If it is not superheated by the throttling calorimeter, the separating calorimeter removes the remaining water. The combination thus makes it possible to determine the quality of a sample with any initial quality.

172. Determination of Engine Performance. Engines are generally purchased on the basis of their thermal performance, expressed as pounds of steam of a given character

which will be required per unit of output. The unit of output is sometimes taken as the indicated horse-power and sometimes as the delivered horse-power. The latter is the more logical since the delivered power is the useful output of the engine.

The steam supplied can be determined easily when the engine exhausts into a surface condenser because under such circumstances all the steam exhausted is condensed to liquid water and can be weighed. When an engine is not fitted with a surface condenser the accurate determination of the steam consumption is more difficult.

The most common method is to isolate the necessary number of boilers and to supply all steam delivered by them to the engine under test. The feed water going to these boilers is then weighed, all losses through blowoff valves and other connections, safety valves, steam separators, etc., are deducted and the remainder is taken as the steam supplied the engine.

Occasionally steam meters are used but such devices are not yet developed to the point where they can be regarded as at all comparable with tank and scales for accuracy.

The character of steam delivered can be determined by pressure gage and thermometer, or steam calorimeter, at the engine.

The indicated horse-power can be determined by means of the steam engine indicator already described in Chapter VIII.

The determination of the delivered or brake horse-power is not always as simple a matter as might be desired. There are numerous ways available but many of them are cumbersome, or inaccurate or difficult of application, or have some other drawback. Some of the more common are considered in the following paragraphs.

173. Determination of Delivered Horse-power. The devices used to determine delivered horse-power are all classifiable as dynamometers. There are two distinctly

different types known respectively as *absorption dynamometers* and *transmission dynamometers.*

The absorption dynamometer absorbs the energy which it measures as is indicated by its name. The transmission dynamometer merely measures the energy as it passes or is transmitted without absorbing any of it, except a negligible amount in its own friction.

One of the most common forms of absorption dynamometer is a device known as the *Prony brake.* It is probable that the term brake horse-power comes from the use of such

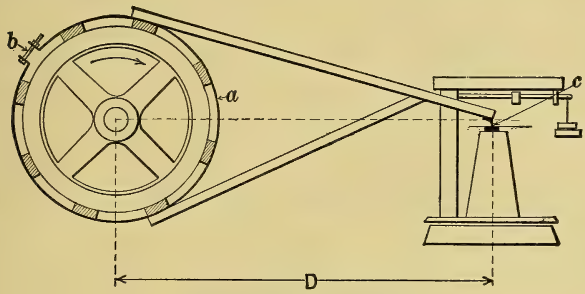


FIG. 251.—Prony Brake.

a brake for determining the delivered horse-power of engines many years ago. One type of Prony brake as applied to a steam engine is shown semi-diagrammatically in Fig. 251.

In this figure the wheel represents either the fly wheel of the engine, a belt wheel or the engine shaft or a special wheel fastened to the shaft for the purposes of the test. A strap *a* made of steel or leather or any other convenient material surrounds the wheel and is spaced away from its face at intervals by means of blocks of wood or other appropriate material. A clamping device of some convenient form such as that shown at *b* is used to pull the blocks against the face of the wheel with any desired force.

If the wheel is rotated and such a band without the long arm shown in the figure is clamped to it, the band will

rotate with the wheel. If now the band is held stationary while the wheel is rotated the friction of the blocks on the face of the wheel will offer some resistance to the rotation of the wheel. In fact, power will be expended in overcoming this friction and the energy "consumed" will appear as heat. By clamping more or less tightly more or less power can be absorbed in this way. This part of the Prony brake is therefore a power absorber.

The frictional resistance to motion is equivalent to a tangential force applied at the rim of the wheel in such a direction as to resist its motion. If the wheel moves it must exert a tangential force equal and opposite to this resisting force. The energy supplied by the wheel (and absorbed by the brake) is equal to the product of this force by the distance through which it travels. That is

$$\text{Energy in foot pounds per minute} = F \times 2\pi R n, \quad . \quad (107)$$

in which

F = tangential force in pounds;
R = radius of wheel in feet:
n = revolutions per minute.

And the power supplied by the wheel would be

$$\text{Power in h.p.} = \frac{F \times 2\pi R n}{33{,}000}. \quad . \quad . \quad . \quad (108)$$

It thus appears that if the tangential force could be measured the horse-power output could be determined. The tangential force is measured indirectly by measuring the reaction at the knife edge *c* at the end of the arm which holds the band or brake against rotation. It is obvious that the reaction at the knife edge which may be called F' must be as much smaller than the tangential force F required to hold the

brake stationary if applied at the surface of the wheel, as the distance D is greater than the radius R. That is

$$\frac{F'}{F}=\frac{R}{D},$$

and

$$F'D=FR.$$

The product $F'D$ may therefore be substituted in eq. (108) for FR giving

$$\text{Horse-power absorbed by brake}=\frac{F'\times 2\pi Dn}{33{,}000}, \quad . \quad (109)$$

which is the equation for the Prony brake.

The force or reaction F' is measured by means of scales as shown. It is the total reading of the scales when the wheel is revolving, less that part of the reading which is due to the weight of the brake arm and the weight of the pedestal on which the brake arm acts.

The Prony brake cannot be used to absorb great quantities of energy. It can be designed to care for several hundred horse-power but becomes cumbersome when built to absorb more than about 100 horse-power.

There are many other forms of brake and many of them can be used to absorb large quantities of energy. A brake known as the Alden brake is shown diagrammatically in Fig. 252.

The disk a is keyed to the shaft and rotated by the engine under test. The spaces b are filled with water under any desired pressure and press flexible diaphragms c against the rotating disk. The friction between these diaphragms and the disk absorbs the power generated by the engine.

The casing d to which the diaphragms are fastened is free to rotate and is prevented from doing so by weights suspended from an arm or by resting the end of the arm

on scales as previously described. The formula is the same as for the Prony brake.

A brake of this type can be made up with numerous

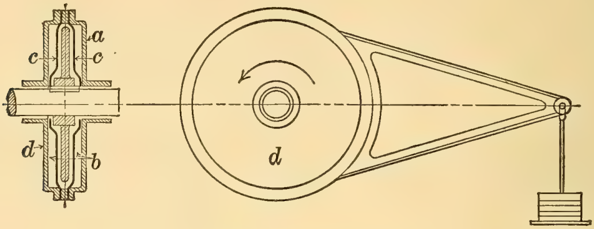


Fig. 252.—Alden Brake.

rotating disks and can thus be made to absorb large amounts of power.

One of the most common forms of absorption dynamometers is a *direct connected electric generator*, that is, a generator which is coupled to the shaft of the engine under test. If all losses in the generator are added to its electrical output the resultant sum must be the input to the generator and the output from the engine. The use of generators in this way makes possible the convenient testing of large engines and turbines which are used to drive direct connected generators.

Transmission dynamometers merely pass the energy along and measure it while doing so instead of absorbing it as in the case of absorption dynamometers. Many transmission dynamometers have been developed but very few have been widely used for measuring the output of steam engines. One has, however, achieved some prominence in this use because it happens to offer about the only available means of determining the output of a marine engine or marine turbine in place in the ship and driving a propeller.

This form is called a torsion dynamometer because its operation depends on the torsion or twist in a long shaft delivering power. Several forms are available but all

merely indicate or record the relative twist between points on the shaft surface at a known distance apart along the length of the shaft. The horse-power transmitted is then calculated by formulas involving the shaft diameter, physical constants of the shaft material, angle of twist, revolutions and length of shaft under test.

TABLES

TABLE GIVING THE PROPERTIES OF ONE POUND OF SATURATED STEAM

[Condensed and rearranged from Marks and Davis's STEAM TABLES AND DIAGRAMS, 1909, by permission of the publishers, Longmans, Green & Co., with addition of lines and columns as indicated by asterisks (*). The first eleven lines constitute a table based on temperatures which are so selected as to give (approximately) intervals of half an inch of Hg, in column 1, without using fractional degrees. The rest of the table is based on absolute pressures. At the heads of columns are given the alternative symbols which are often used.]

Pressure.		Temp. ° F.	Heat—British Thermal Units.					Entropy.			Sp. Vol. Cu.ft.	Pres. Abs. Lbs. Sq.in
Inches of Mercury.	Abs. Lbs. Sq.in.		Total.	Water.	Latent. Total.	Latent. Internal.	Latent. External.	Total.	Liquid.	Vapor.		
	p	t	$\lambda(H)$	$q(h)$	$r(L)$	$\rho(I)$	APu*	$\Delta\phi(N)$	$\Delta\phi_e(n)(\theta)$	$\Delta\phi_v\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	$V(v)(s)$	p
0.1804	0.0886	**32**	1073.4	0.0	1073.4	1019.3	54.1	2.1832	0.0000	2.1832	3294.0	
0.50+	0.2472	**59**	1085.4	27.08	1058.3	1001.3	57.0	2.0944	0.0536	2.0408	1249.2	
1.00−	0.4893	**79**	1094.3	47.04	1047.3	988.1	59.2	2.0358	0.0913	1.9445	657.0	
1.50+	0.741	**92**	1100.1	60.0	1040.0	979.4	60.6	2.0007	0.1151	1.8856	442.2	
2.00−	0.975	**101**	1104.0	68.97	1035.1	973.5	61.6	1.9776	0.1313	1.8463	341.0	
2.50+	1.235	**109**	1107.5	76.94	1030.6	968.2	62.4	1.9678	0.1454	1.8124	272.9	
3.00−	1.467	**115**	1110.2	82.92	1027.2	964.2	63.0	1.9435	0.1559	1.7876	231.9	
3.50+	1.736	**121**	1112.8	88.91	1023.9	960.2	63.7	1.9296	0.1662	1.7634	197.9	
4.00+	1.992	**126**	1115.0	93.90	1021.1	956.8	64.3	1.9183	0.1747	1.7436	173.9	
4.50+	2.219	**130**	1116.7	97.89	1018.8	954.1	64.7	1.9095	0.1816	1.7279	157.1	
5.00+	2.467	**134**	1118.4	101.88	1016.5	951.4	65.1	1.9008	0.1883	1.7125	142.2	

	1	101.83	1104.4	69.8	1034.6	972.9	61.7	1.9754	0.1327	1.8427	333.0	1
	2	126.15	1115.0	94.0	1021.0	956.7	64.3	1.9180	0.1749	1.7431	173.5	2
	3	141.52	1121.6	109.4	1012.3	946.4	65.9	1.8848	0.2008	1.6840	118.5	3
	4	153.01	1126.5	120.9	1005.7	938.6	67.1	1.8614	0.2198	1.6416	90.5	4
	5	162.28	1130.5	130.1	1000.3	932.4	67.9	1.8432	0.2348	1.6084	73.33	5
	6	170.06	1133.7	137.9	995.8	927.0	68.8	1.8285	0.2471	1.5814	61.89	6
	7	176.85	1136.5	144.7	991.8	922.4	69.4	1.8161	0.2579	1.5582	53.56	7
	8	182.86	1139.0	150.8	988.2	918.2	70.0	1.8053	0.2673	1.5380	47.27	8
	9	188.27	1141.1	156.2	985.0	914.4	70.6	1.7958	0.2756	1.5202	42.36	9
	10	193.22	1143.1	161.1	982.0	910.9	71.1	1.7874	0.2832	1.5042	38.38	10
	11	197.75	1144.9	165.7	979.2	907.8	71.4	1.7797	0.2902	1.4895	35.10	11
	12	201.96	1146.5	169.9	976.6	904.8	71.8	1.7727	0.2967	1.4760	32.36	12
	13	205.87	1148.0	173.8	974.2	902.0	72.2	1.7664	0.3025	1.4639	30.03	13
	14	209.55	1149.4	177.5	971.9	899.3	72.6	1.7604	0.3081	1.4523	28.02	14
*0	14.7	212.00	1150.4	180.0	970.4	897.6	72.8	1.7565	0.3118	1.4447	26.79	14.7
0.3	15	213.0	1150.7	181.0	969.7	896.8	72.9	1.7549	0.3133	1.4416	26.27	15
1.3	16	216.3	1152.0	184.4	967.6	894.4	73.2	1.7494	0.3183	1.4311	24.79	16
2.3	17	219.4	1153.1	187.5	965.6	892.1	73.5	1.7444	0.3229	1.4215	23.38	17
3.3	18	222.4	1154.2	190.5	963.7	889.9	73.8	1.7400	0.3273	1.4127	22.16	18
4.3	19	225.2	1155.2	193.4	961.8	887.8	74.0	1.7360	0.3315	1.4045	21.07	19
5.3	20	228.0	1156.2	196.1	960.0	885.8	74.2	1.7320	0.3355	1.3965	20.08	20
6.3	21	230.6	1157.1	198.8	958.3	883.9	74.4	1.7280	0.3393	1.3887	19.18	21
7.3	22	233.1	1158.0	201.3	956.7	882.0	74.7	1.7241	0.3430	1.3811	18.37	22
8.3	23	235.5	1158.8	203.8	955.1	880.2	74.9	1.7204	0.3465	1.3739	17.62	23
9.3	24	237.8	1159.6	206.1	953.5	878.5	75.0	1.7169	0.3499	1.3670	16.93	24
10.3	25	240.1	1160.4	208.4	952.0	876.8	75.2	1.7136	0.3532	1.3604	16.30	25

SATURATED STEAM *(Continued)*

Pressure.		Temp. ° F.	Heat—British Thermal Units.					Entropy.			Sp. Vol. Cu.ft.	Pres. Abs. Lbs. Sq.in.
* Gauge Pressure. Lbs. Sq.in.	Abs. Lbs. Sq.in.		Total.	Water.	Latent. Total.	Latent. Internal.	Latent. External.	Total.	Liquid.	Vapor.		
	p	t	$\lambda(H)$	$q(h)$	$r(L)$	$\rho(I)$	APu*	$\Delta\phi(N)$	$\Delta\phi_e(n)(\theta)$	$\Delta\phi_v\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	$V(v)(s)$	p
11.3	**26**	242.2	1161.2	210.6	950.6	875.1	75.5	1.7106	0.3564	1.3542	15.72	**26**
12.3	**27**	244.4	1161.9	212.7	949.2	873.5	75.7	1.7077	0.3594	1.3483	15.18	**27**
13.3	**28**	246.4	1162.6	214.8	947.8	872.0	75.8	1.7048	0.3623	1.3425	14.67	**28**
14.3	**29**	248.4	1163.2	216.8	946.4	870.5	75.9	1.7019	0.3652	1.3367	14.19	**29**
15.3	**30**	250.3	1163.9	218.8	945.1	869.0	76.1	1.6991	0.3680	1.3311	13.74	**30**
17.3	**32**	254.1	1165.1	222.6	942.5	866.2	76.3	1.6938	0.3733	1.3205	12.93	**32**
19.3	**34**	257.6	1166.3	226.2	940.1	863.4	76.7	1.6891	0.3784	1.3107	12.22	**34**
21.3	**36**	261.0	1167.3	229.6	937.7	860.8	76.9	1.6846	0.3832	1.3014	11.58	**36**
23.3	**38**	264.2	1168.4	232.9	935.5	858.3	77.2	1.6802	0.3877	1.2925	11.01	**38**
25.3	**40**	267.3	1169.4	236.1	933.3	855.9	77.4	1.6761	0.3920	1.2841	10.49	**40**
27.3	**42**	270.2	1170.3	239.1	931.2	853.6	77.6	1.6721	0.3962	1.2759	10.02	**42**
29.3	**44**	273.1	1171.2	242.0	929.2	851.3	77.9	1.6683	0.4002	1.2681	9.59	**44**
31.3	**46**	275.8	1172.0	244.8	927.2	849.2	78.0	1.6647	0.4040	1.2607	9.20	**46**
33.3	**48**	278.5	1172.8	247.5	925.3	847.1	78.2	1.6613	0.4077	1.2536	8.84	**48**
35.3	**50**	281.0	1173.6	250.1	923.5	845.0	78.5	1.6581	0.4113	1.2468	8.51	**50**

37.3	**52**	283.5	1174.3	252.6	921.7	843.1	78.6	1.6549	0.4147	1.2402	8.20	**52**
39.3	**54**	285.9	1175.0	255.1	919.9	841.1	78.8	1.6519	0.4180	1.2339	7.91	**54**
41.3	**56**	288.2	1175.7	257.5	918.2	839.3	78.9	1.6490	0.4212	1.2278	7.65	**56**
43.3	**58**	290.5	1176.4	259.8	916.5	837.4	79.1	1.6460	0.4242	1.2218	7.40	**58**
45.3	**60**	292.7	1177.0	262.1	914.9	835.6	79.3	1.6432	0.4272	1.2160	7.17	**60**
47.3	**62**	294.9	1177.6	264.3	913.3	833.9	79.4	1.6406	0.4302	1.2104	6.95	**62**
49.3	**64**	297.0	1178.2	266.4	911.8	832.2	79.6	1.6380	0.4330	1.2050	6.75	**64**
51.3	**66**	299.0	1178.8	268.5	910.2	830.5	79.7	1.6355	0.4358	1.2007	6.56	**66**
53.3	**68**	301.0	1179.3	270.6	908.7	828.9	79.8	1.6331	0.4385	1.1946	6.38	**68**
55.3	**70**	302.9	1179.8	272.6	907.2	827.3	79.9	1.6307	0.4411	1.1896	6.20	**70**
57.3	**72**	304.8	1180.4	274.5	905.8	825.8	80.0	1.6285	0.4437	1.1848	6.04	**72**
59.3	**74**	306.7	1180.9	276.5	904.4	824.2	80.2	1.6263	0.4462	1.1810	5.89	**74**
61.3	**76**	308.5	1181.4	278.3	903.0	822.7	80.3	1.6242	0.4487	1.1755	5.74	**76**
63.3	**78**	310.3	1181.8	280.2	901.7	821.3	80.4	1.6221	0.4511	1.1710	5.60	**78**
65.3	**80**	312.0	1182.3	282.0	900.3	819.8	80.5	1.6200	0.4535	1.1665	5.47	**80**
67.3	**82**	313.8	1182.8	283.8	899.0	818.4	80.6	1.6180	0.4557	1.1623	5.34	**82**
69.3	**84**	315.4	1183.2	285.5	897.7	817.0	80.7	1.6160	0.4579	1.1581	5.22	**84**
71.3	**86**	317.1	1183.6	287.2	896.4	815.6	80.8	1.6141	0.4601	1.1540	5.10	**86**
73.3	**88**	318.7	1184.0	288.9	895.2	814.3	80.9	1.6123	0.4623	1.1500	5.00	**88**
75.5	**90**	320.3	1184.4	290.5	893.9	813.0	80.9	1.6105	0.4644	1.1461	4.89	**90**
77.3	**92**	321.8	1184.8	292.1	892.7	811.7	81.0	1.6087	0.4664	1.1423	4.79	**92**
79.3	**94**	323.4	1185.2	293.7	891.5	810.4	81.1	1.6069	0.4684	1.1385	4.69	**94**
81.3	**96**	324.9	1185.6	295.3	890.3	809.1	81.2	1.6052	0.4704	1.1348	4.60	**96**
83.3	**98**	326.4	1186.0	296.8	889.2	807.9	81.3	1.6036	0.4724	1.1312	4.51	**98**
85.3	**100**	327.8	1186.3	298.3	888.0	806.6	81.4	1.6020	0.4743	1.1277	4.429	**100**

SATURATED STEAM (*Continued*)

Pressure.		Temp. ° F.	Heat—British Thermal Units.					Entropy.			Sp. Vol. Cu.ft.	Pres. Abs. Lbs. Sq.in.
* Gauge Pressure. Lbs. Sq.in.	Abs. Lbs. Sq.in.		Total.	Water.	Latent. Total.	Latent. Internal.	Latent. External.	Total.	Liquid.	Vapor.		
	p	t	$\lambda(H)$	$q(h)$	$r(L)$	$\rho(I)$	APu*	$\Delta\phi(N)$	$\Delta\phi_e(n)(\theta)$	$\Delta\phi_v\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	$V(v)(s)$	p
90.3	**105**	331.4	1187.2	302.0	885.2	803.6	81.6	1.5980	0.4789	1.1191	4.230	**105**
95.3	**110**	334.8	1188.0	305.5	882.5	800.7	81.8	1.5942	0.4834	1.1108	4.047	**110**
100.3	**115**	338.1	1188.8	309.0	879.8	797.9	81.9	1.5907	0.4877	1.1030	3.880	**115**
105.3	**120**	341.3	1189.6	312.3	877.2	795.2	82.0	1.5873	0.4919	1.0954	3.726	**120**
110.3	**125**	344.4	1190.3	315.5	874.7	792.6	82.1	1.5839	0.4959	1.0880	3.583	**125**
115.3	**130**	347.4	1191.0	318.6	872.3	790.0	82.3	1.5807	0.4998	1.0809	3.452	**130**
120.3	**135**	350.3	1191.6	321.7	869.9	787.5	82.4	1.5777	0.5035	1.0742	3.331	**135**
125.3	**140**	353.1	1192.2	324.6	867.6	785.0	82.6	1.5747	0.5072	1.0675	3.219	**140**
130.3	**145**	355.8	1192.8	327.4	865.4	782.7	82.7	1.5719	0.5107	1.0612	3.112	**145**
135.3	**150**	358.5	1193.4	330.2	863.2	780.4	82.8	1.5692	0.5142	1.0550	3.012	**150**
140.3	**155**	361.0	1194.0	332.9	861.0	778.1	82.9	1.5664	0.5175	1.0489	2.920	**155**
145.3	**160**	363.6	1194.5	335.6	858.8	775.8	83.0	1.5639	0.5208	1.0431	2.834	**160**
150.3	**165**	366.0	1195.0	338.2	856.8	773.6	83.2	1.5615	0.5239	1.0376	2.753	**165**
155.3	**170**	368.5	1195.4	340.7	854.7	771.5	83.2	1.5590	0.5269	1.0321	2.675	**170**
160.3	**175**	370.8	1195.9	343.2	852.7	769.4	83.3	1.5567	0.5299	1.0268	2.602	**175**

165.3	**180**	373.1	1196.4	345.6	850.8	767.4	83.4	1.5543	0.5328	1.0215	2.533	**180**
170.3	**185**	375.4	1196.8	348.0	848.8	765.4	83.4	1.5520	0.5356	1.0164	2.468	**185**
175.3	**190**	377.6	1197.3	350.4	846.9	763.4	83.5	1.5498	0.5384	1.0114	2.406	**190**
180.3	**195**	379.8	1197.7	352.7	845.0	761.4	83.6	1.5476	0.5410	1.0066	2.346	**195**
185.3	**200**	381.9	1198.1	354.9	843.2	759.5	83.7	1.5456	0.5437	1.0019	2.290	**200**
235.3	**250**	401.1	1201.5	375.2	826.3	742.0	84.3	1.5276	0.5676	0.9600	1.850	**250**
285.3	**300**	417.5	1204.1	392.7	811.3	726.8	84.5	1.5129	0.5878	0.9251	1.551	**300**
385.3	**400**	444.8	1208	422	786.0	701	85	1.489	0.621	0.868	1.17	**400**
485.3	**500**	467.3	1210	448	762	678	84	1.470	0.648	0.822	0.93	**500**
*985.3	**1000**	545	1200	535	665	587	78	1.39	0.73	0.66	0.4	**1000**
*	**2000**	640	?	?	490	430	60	?	?	0.45	0.2	**2000**
*Crit.	**3200**	706	?	?	0	0	0	?	?	?	0.05	**3200**

PROPERTIES OF ONE POUND OF SUPERHEATED STEAM

[Condensed from Marks and Davis's Steam Tables and Diagrams, 1909, by permission of the publishers, Longmans, Green & Co.]

Sp.V. = specific volume in cu.ft.; ΔQ = B.t.u. total heat above 32° F.; $\Delta\phi$ = total entropy above 32° F.

Absolute Pressure. Lbs. Sq.in. (Sat. Temp. ° F.)		Degrees of Superheat.						
		0	50	100	150	200	250	300
15 (213)	Sp. V.	26.27	28.40	30.46	32.50	34.53	36.56	38.58
	ΔQ	1150.7	1174.2	1197.6	1221.0	1244.4	1267.7	1291.1
	$\Delta\phi$	1.7549	1.7886	1.8199	1.8492	1.8768	1.9029	1.9276
50 (281)	Sp. V.	8.51	9.19	9.84	10.48	11.11	11.74	12.36
	ΔQ	1173.6	1198.8	1223.4	1247.7	1271.8	1295.8	1319.7
	$\Delta\phi$	1.6581	1.6909	1.7211	1.7491	1.7755	1.8002	1.8237
100 (327.8)	Sp. V.	4.43	4.79	5.14	5.47	5.80	6.12	6.44
	ΔQ	1186.3	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8
	$\Delta\phi$	1.6020	1.6358	1.6658	1.6933	1.7188	1.7428	1.7656
110 (334.8)	Sp. V.	4.05	4.38	4.70	5.01	5.31	5.61	5.90
	ΔQ	1188.0	1215.9	1242.0	1267.1	1291.9	1316.2	1340.4
	$\Delta\phi$	1.5942	1.6282	1.6583	1.6857	1.7110	1.7350	1.7576
120 (341.3)	Sp. V.	3.73	4.04	4.33	4.62	4.89	5.17	5.44
	ΔQ	1189.6	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7
	$\Delta\phi$	1.5873	1.6216	1.6517	1.6789	1.7041	1.7280	1.7505
130 (347.4)	Sp. V.	3.45	3.74	4.02	4.28	4.54	4.80	5.05
	ΔQ	1191.0	1219.7	1246.1	1271.4	1296.2	1320.6	1344.9
	$\Delta\phi$	1.5807	1.6153	1.6453	1.6724	1.6976	1.7213	1.7437
140 (353.1)	Sp. V.	3.22	3.49	3.75	4.00	4.24	4.48	4.71
	ΔQ	1192.2	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9
	$\Delta\phi$	1.5747	1.6096	1.6395	1.6666	1.6916	1.7152	1.7376
150 (358.5)	Sp. V.	3.01	3.27	3.51	3.75	3.97	4.19	4.41
	ΔQ	1193.4	1223.0	1249.6	1275.1	1300.0	1324.5	1348.8
	$\Delta\phi$	1.5692	1.6043	1.6343	1.6612	1.6862	1.7097	1.7320

PROPERTIES OF ONE POUND OF SUPERHEATED STEAM
(Continued)

Absolute Pressure. Lbs. Sq.in. / Sat. Temp. ° F.		Degrees of Superheat.						
		0	50	100	150	200	250	300
160 (363.6)	Sp. V.	2.83	3.07	3.30	3.53	3.74	3.95	4.15
	ΔQ	1194.5	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6
	$\Delta \phi$	1.5693	1.5993	1.6292	1.6561	1.6810	1.7043	1.7266
170 (368.5)	Sp. V.	2.68	2.91	3.12	3.34	3.54	3.73	3.92
	ΔQ	1195.4	1225.9	1252.8	1278.4	1303.3	1327.9	1352.3
	$\Delta \phi$	1.5590	1.5947	1.6246	1.6513	1.6762	1.6994	1.7217
180 (373.1)	Sp. V.	2.53	2.75	2.96	3.16	3.35	3.54	3.72
	ΔQ	1196.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9
	$\Delta \phi$	1.5543	1.5904	1.6201	1.6468	1.6716	1.6948	1.7169
190 (377.6)	Sp. V.	2.41	2.62	2.81	3.00	3.19	3.37	3.55
	ΔQ	1197.3	1228.6	1255.7	1281.3	1306.3	1330.9	1355.5
	$\Delta \phi$	1.5498	1.5862	1.6159	1.6425	1.6627	1.6904	1.7124
200 (381.9)	Sp. V.	2.29	2.49	2.68	2.86	3.04	3.21	3.38
	ΔQ	1198.1	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0
	$\Delta \phi$	1.5456	1.5823	1.6120	1.6385	1.6632	1.6862	1.7082
300 (417.5)	Sp. V.	1.55	1.69	1.83	1.96	2.09	2.21	2.33
	ΔQ	1204.1	1240.3	1268.2	1294.0	1319.3	1344.3	1369.2
	$\Delta \phi$	1.5129	1.5530	1.5824	1.6082	1.6323	1.6550	1.6765
500 (467.3)	Sp. V.	0.93	1.03	1.11	1.22	1.31	1.39	1.47
	ΔQ	1210.0	1256	1285	1311	1337	1362	1388
	$\Delta \phi$	1.470	1.519	1.548	1.573	1.597	1.619	1.640

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